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WESTINGHOUSE GEARED TURBINE PROPELLING MACHINERY FOR SWEDISH BATTLESHIPS *GUSTAF V* AND *DROTTNING VICTORIA.*

BY WILLIAM W. SMITH, MEMBER.

The latest developments in geared turbines for the propulsion of naval vessels is represented by the Westinghouse geared turbine machinery which is being supplied for Swedish Battleships *Gustaf V* and *Drottning Victoria.* These vessels are being constructed at Kockums Met. Verkstads Aktiebolag, in Malmö, and Gotaverken in Göthenburg, Sweden, respectively, and the turbines are being manufactured by the Motala Verkstads Nya Aktiebolag, in Motala, Sweden, to design prepared by the Westinghouse Machine Company. The principal data for the vessels are:

Length, feet	295¾
Beam, feet	64
Draft, feet	20½
Displacement, tons	7,000
Speed, knots	22.5
S. H. P.	22,000
Propellers, number	2

Armament: Four 11-inch, eight 6-inch, six 12-pounders, and two submerged torpedo tubes.

Complement, 408.

PROPELLING MACHINERY.

The propelling machinery is of the Westinghouse geared turbine type and is composed of two units located in engine rooms amidships below the protective deck. Each unit consists of two turbines of the divided-flow type, which drive the propeller through a double-pinion reduction gear as shown in Figs. 1 to 4.

The propelling machinery is designed to develop 22,000 S.H.P. and to give a speed of 22.5 knots. The steam consumption guarantees for the turbines are exceptionally good, and are far superior to the results which have been obtained with the types of machinery now in use. The performance data is as follows:

Speed (approximate)	22.5	20	12.5
R.P.M., propeller	200	177	115
R.P.M., turbine	3,600	3,186	2,070
S.H.P.	22,000	15,000	3,150
Water rate, lbs. per S.H.P. at propeller shaft	10.9	11.1	13.3

The astern turbines will develop 41 per cent. of the ahead power with the same flow of steam, or 9,000 S.H.P. at full power.

The operating conditions for the above are to be as follows:

Steam pressure at turbine	265 pounds gage.
Quality	Dry and Saturated.
Vacuum in exhaust	28.27 of mercury referred to a 30 in. barometer.

The machinery weights per shaft are as follows:

2 Turbines, pounds	47,800
1 Reduction gear, pounds	84,500
Total, pounds	132,300
Total tons (2,240 lbs.)	59.1
Weight per S.H.P.	12.0

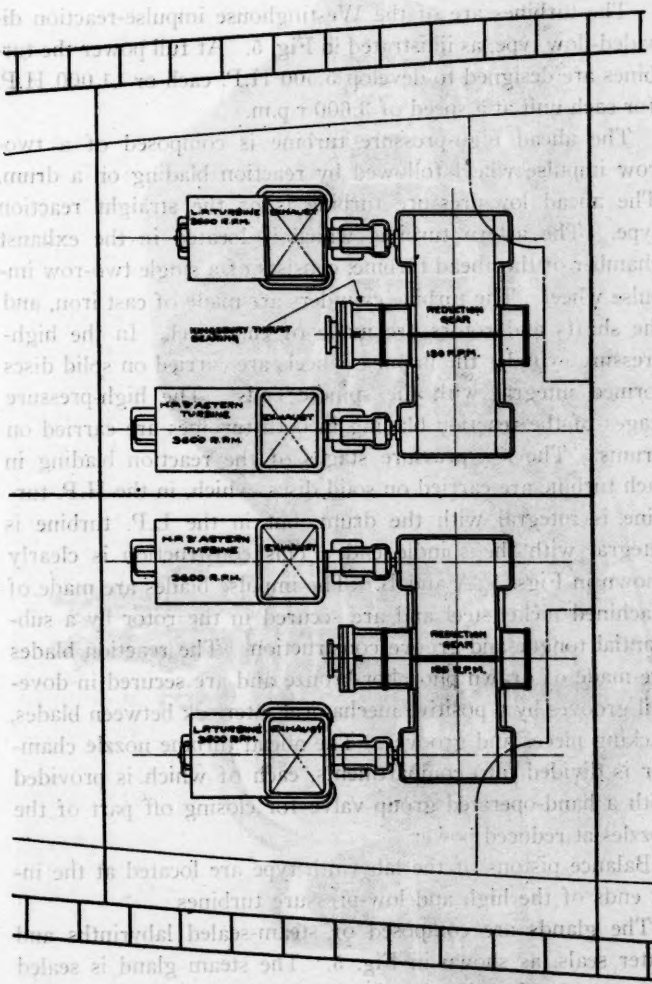


FIG. 1.—PLAN AND ELEVATIONS OF ENGINE ROOMS SHOWING ARRANGEMENT OF PROPELLING MACHINERY.

TURBINES.

The turbines are of the Westinghouse impulse-reaction divided-flow type, as illustrated in Fig. 5. At full power the turbines are designed to develop 5,500 H.P. each or 11,000 H.P. for each unit at a speed of 3,600 r.p.m.

The ahead high-pressure turbine is composed of a two-row impulse wheel followed by reaction blading on a drum. The ahead low-pressure turbine is of the straight reaction type. The astern turbine, which is located in the exhaust chamber of the ahead turbine, consists of a single two-row impulse wheel. The turbine cylinders are made of cast iron, and the shafts and rotors are made of cast steel. In the high-pressure cylinder the impulse wheels are carried on solid discs formed integral with the spindle ends. The high-pressure stages of the reaction blading in both turbines are carried on drums. The low-pressure stages of the reaction blading in each turbine are carried on solid discs, which, in the H.P. turbine is integral with the drum, but in the L.P. turbine is integral with the spindle end. This construction is clearly shown in Figs. 5, A and B. The impulse blades are made of machined nickel-steel and are secured in the rotor by a substantial tongue-and-groove construction. The reaction blades are made of drawn phosphor-bronze and are secured in dovetail grooves by a positive mechanical interlock between blades, packing pieces and grooves. The ahead turbine nozzle chamber is divided into compartments, each of which is provided with a hand-operated group valve for closing off part of the nozzles at reduced power.

Balance pistons of the labyrinth type are located at the inlet ends of the high and low-pressure turbines.

The glands are composed of steam-sealed labyrinths and water seals, as shown in Fig. 5. The steam gland is sealed from an annular chamber in the usual manner. The water seal operates on the principle of a centrifugal pump, the impeller being provided with radial paddles or ribs, which revolve in the chamber, supplied with water at the periphery of the im-

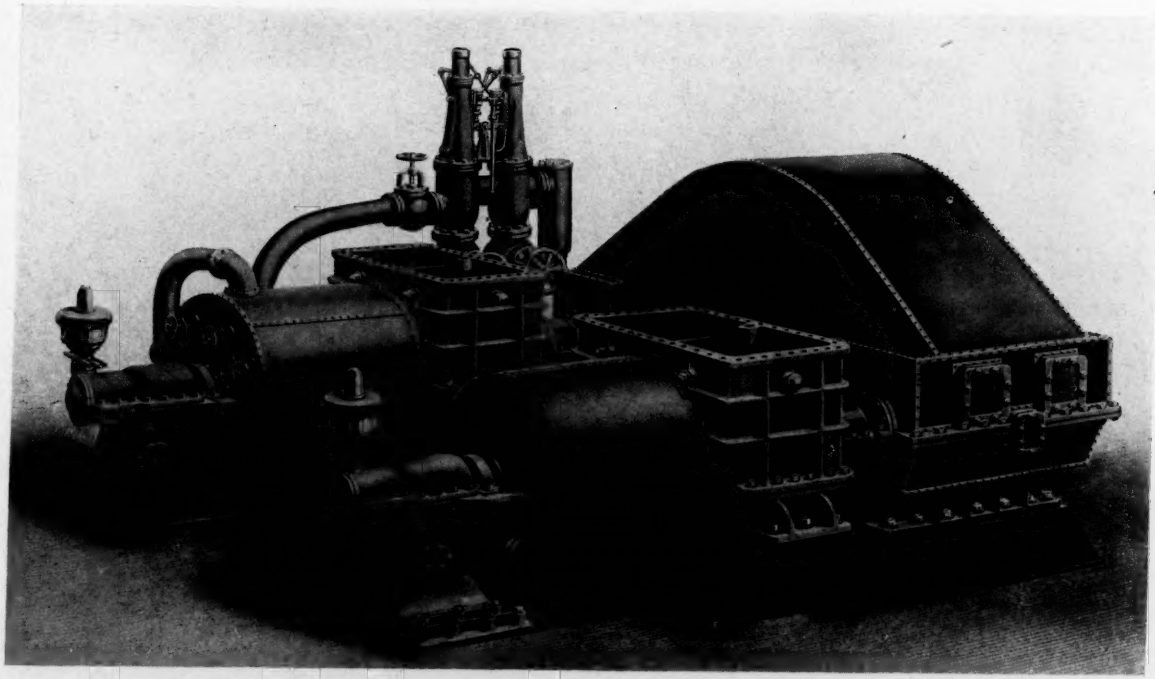


FIG. 2.—PERSPECTIVE VIEW OF COMPLETE GEARED TURBINE PROPELLING UNIT SHOWING AHEAD AND ASTERN THROTTLE VALVES, OVERSPEED GOVERNORS AND NOZZLE GROUP VALVES.

peller. The centrifugal action of the paddles on the water causes it to revolve and to form a perfect seal against the leakage of air. The water seal is designed to operate from full down to half speed. Below the latter speed the centrifugal force is not sufficient to maintain the seal, and, therefore, from stop to half speed it is necessary to use the steam-sealed

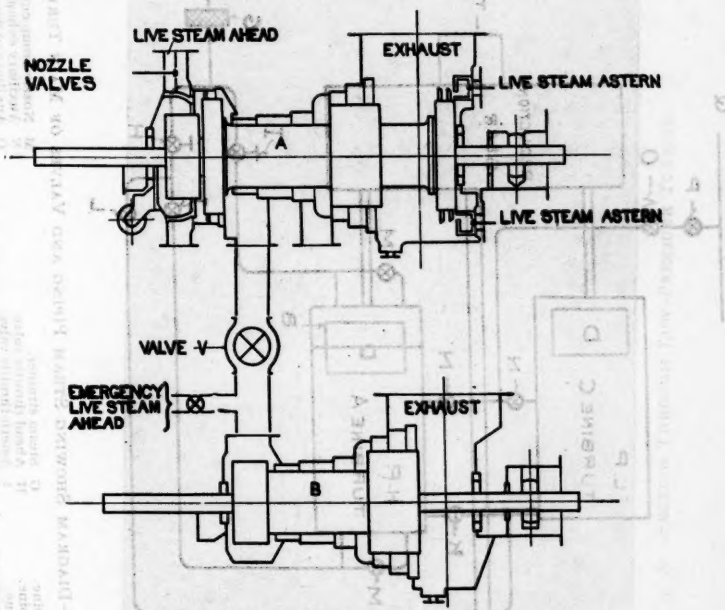


FIG. 3.—DIAGRAM SHOWING PRINCIPLE OF OPERATION, PIPE CONNECTIONS, ETC.

A — High-pressure turbine.
B — Low-pressure turbine.
V — Low-pressure turbine cut-out valve.

labyrinth. Consequently the steam seal will be used when steaming at low speed, maneuvering and standing by.

Steam is supplied to the labyrinth glands at a pressure of about five pounds in the usual manner. Water is supplied to the water glands from a gravity tank provided for this purpose.

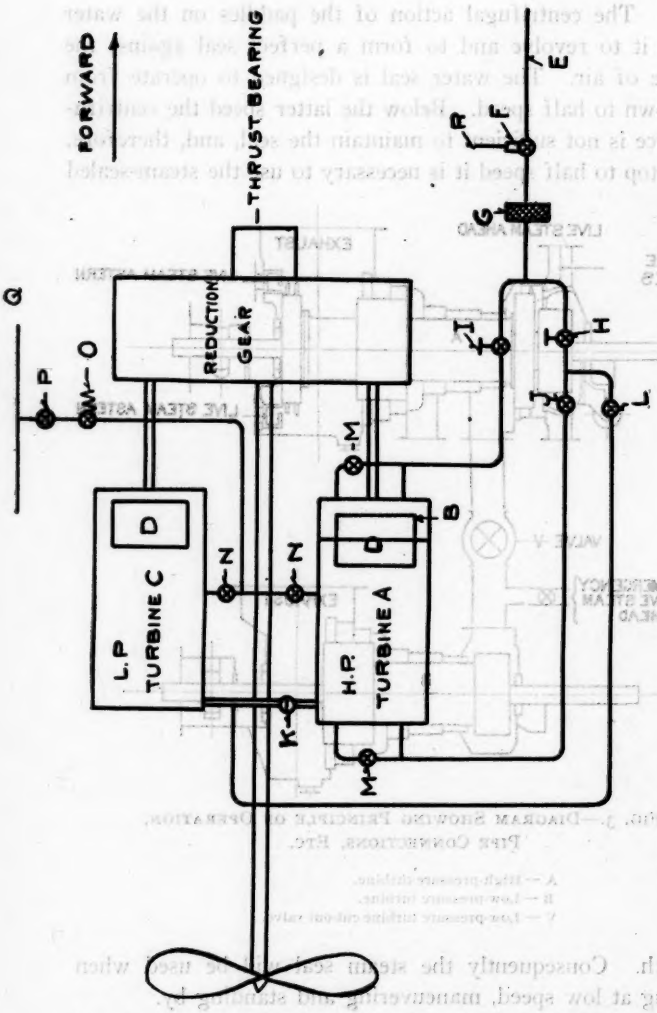


FIG. 4.—DIAGRAM SHOWING STEAM PIPING AND VALVES OF MAIN TURBINES.

A High-pressure ahead turbine.
B High-pressure astern turbine.
C Low-pressure ahead turbine.
D Exhaust connections to condenser.
E Main steam line from boilers.
F Automatic governor overspeed valve.
G Steam strainer.
H Ahead throttle valve.
I Astern throttle valve.
J High-pressure turbine cut-out valve.
K Low-pressure turbine cut-out valve.
L Emergency steam valve to low-pressure turbine.
M Nozzle group cut-out valve.
N Auxiliary exhaust inlet to turbine.
O Auxiliary exhaust constant back pressure valve.
P Auxiliary exhaust stop valve.
Q Auxiliary exhaust line.
R Connection from governor valve to turbine governors.

The main bearings are of the self-aligning type and are provided with pads and liners for adjusting the position of the rotor. The shells are of cast iron and the lining of babbitt.

The thrust bearing is of the Kingsbury segmental type and

E Main steam line from boilers.
F Automatic governor overspeed valve.

K Low-pressure turbine cut-out valve.
L Emergency steam valve to low-pressure turbine.

P Auxiliary exhaust stop valve.
Q Auxiliary exhaust line.
R Connection from governor valve to turbine governors.

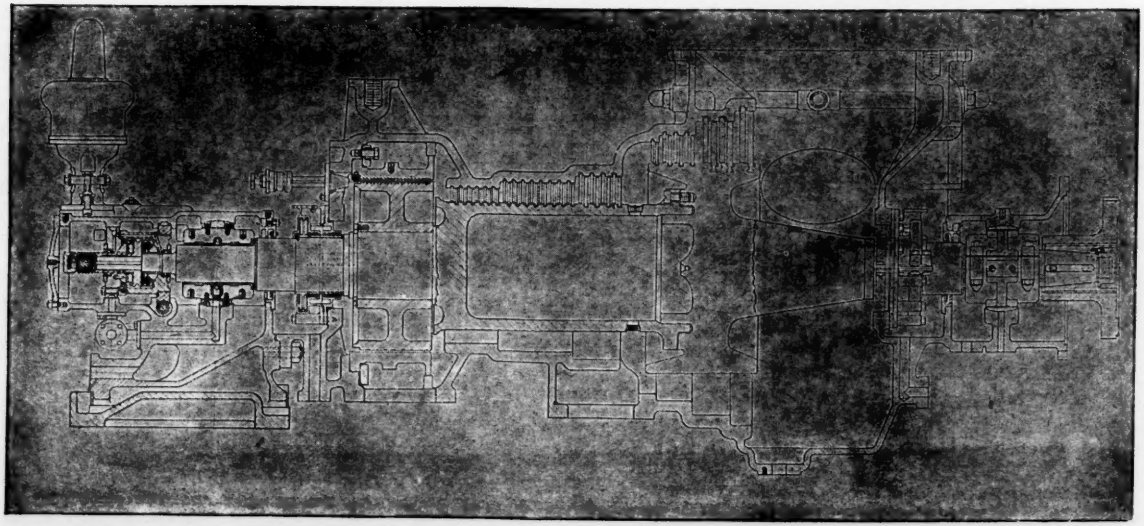


FIG. 5, A.—SECTION THROUGH LOW-PRESSURE TURBINE.

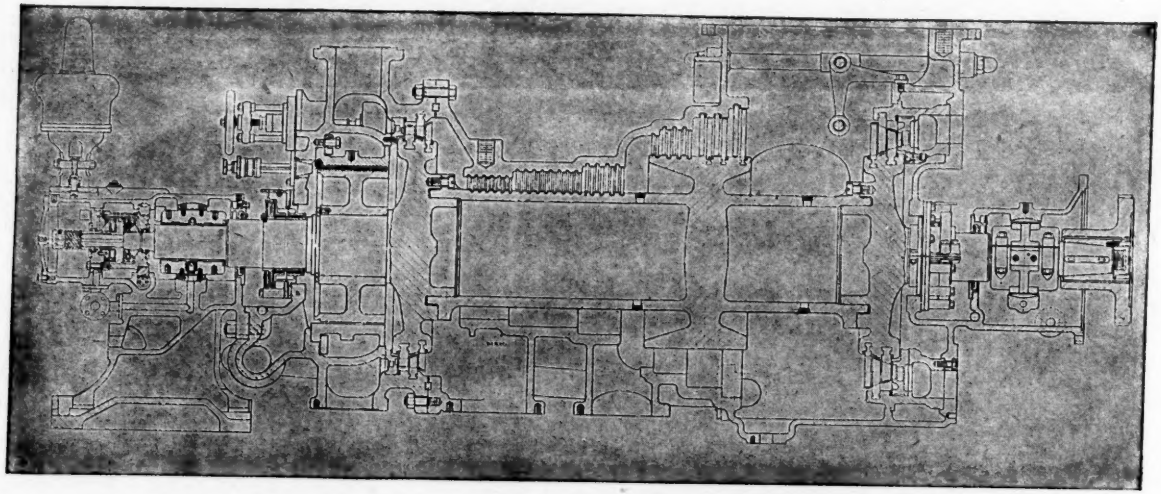


FIG. 5, B.—SECTION THROUGH HIGH-PRESSURE TURBINE AND BACKING TURBINE.

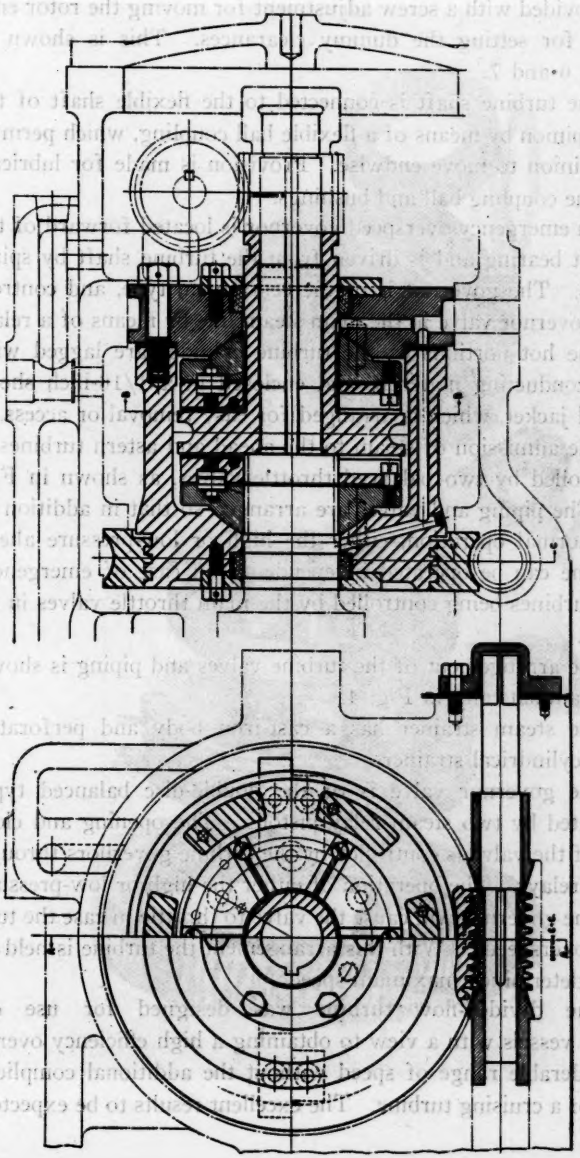


FIG. 6.—SECTION THROUGH TURBINE THRUST BEARING, KINGSBURY TYPE, SHOWING CONSTRUCTION AND METHOD OF ADJUSTMENT.

is provided with a screw adjustment for moving the rotor endwise for setting the dummy clearances. This is shown in Figs. 6 and 7.

The turbine shaft is connected to the flexible shaft of the gear pinion by means of a flexible ball coupling, which permits the pinion to move endwise. Provision is made for lubricating the coupling ball and bushings.

An emergency overspeed governor is located forward of the thrust bearing and is driven from the turbine shaft by spiral gears. The governor is of the centrifugal type, and controls the governor valve in the main steam line by means of a relay.

The hot portions of the turbine cylinder are lagged with non-conducting material, and enclosed in a 1/16-inch sheet-metal jacket, which is arranged for easy removal or access.

The admission of steam to the ahead and astern turbines is controlled by two balanced throttle valves, as shown in Fig. 2. The piping and valves are arranged so that in addition to the normal operation, either the high or low-pressure ahead turbine can be operated independently in case of emergency, the turbines being controlled by the main throttle valves in all cases.

The arrangement of the turbine valves and piping is shown diagrammatically in Fig. 4.

The steam strainer has a cast-iron body and perforated steel cylindrical strainer.

The governor valve is of the double-disc balanced type, operated by two steam relay pistons. The opening and closing of the valve is controlled by the turbine governors through their relays. The operation of either the high or low-pressure turbine governor will cause the valve to throttle in case the turbine overspeeds. With this arrangement the turbine is held at a predetermined maximum speed.

The divided-flow turbine was designed for use on naval vessels with a view to obtaining a high efficiency over a considerable range of speed without the additional complication of a cruising turbine. The excellent results to be expected

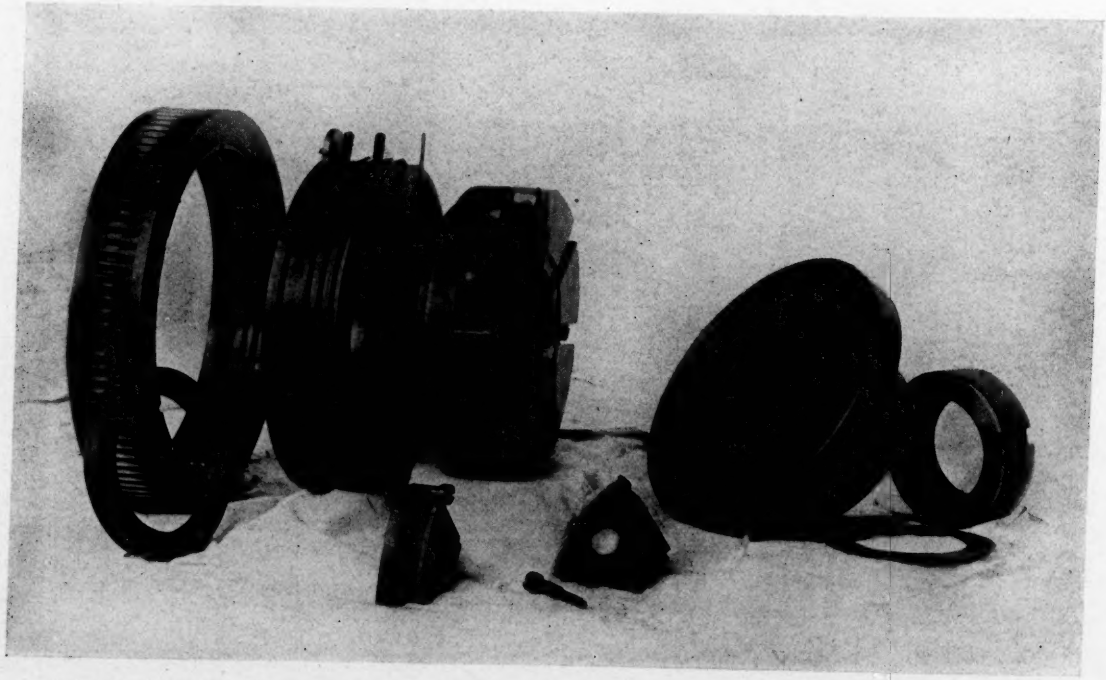


FIG. 7.—VIEW OF KINGSBURY THRUST BEARING FOR TURBINE, SHOWING SEGMENTAL SLIPPERS AND SCREW MECHANISM FOR ADJUSTING ROTOR.

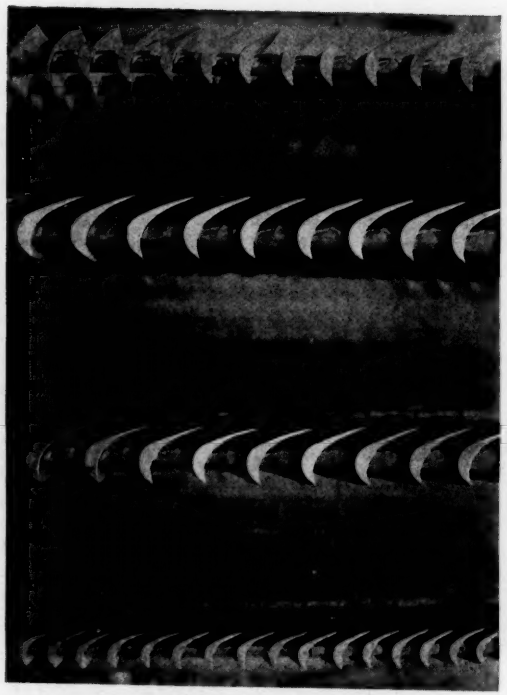


FIG. 9.—VIEW OF TURBINE REACTION BLADING.

with this type of turbine is shown in the water-rate curves, Fig. 13. When steaming at full speed, the steam which is admitted through the high-pressure impulse wheel divides, approximately one-half passing through the high-pressure reaction blading, and the other half through the low-pressure reaction blading. At reduced speeds the low-pressure cut-out valve is closed, causing the steam to flow through the ahead reaction blading only, after having passed through the ahead impulse wheel. This reduction of area for the flow of steam

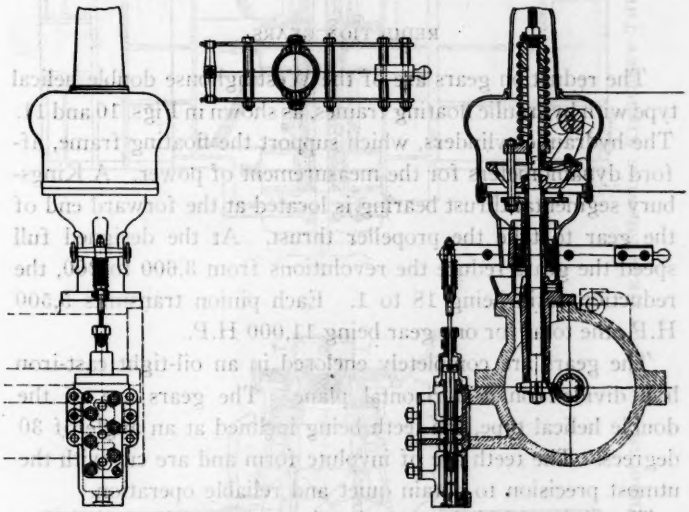


FIG. 8.—SECTION THROUGH TURBINE OVERSPEED GOVERNOR AND RELAY.

improves the pressure distribution and increases the efficiency of the turbine. At reduced or cruising speed the low-pressure turbine revolves idly in a high vacuum, or in case of a long voyage it would be disconnected and would not revolve at all. This arrangement therefore improves the economy at cruising speed, since a small turbine, as required by the speed and power, is used, thus giving a high efficiency at low speed. The blading of this turbine is also designed for efficient operation at the cruising speed, extra rows being provided. Since the

low-pressure turbine is only used at approximately full speed, its blading is arranged for this condition only, less rows being required than would ordinarily be the case. This design also permits a higher speed of rotation, thereby reducing the weights considerably. Since in emergency either turbine can be used independently of the other, complete breakdown is most improbable. Also, since the turbines have a great overload capacity, it would be possible in case of emergency to develop three-fourths or more power with one turbine only.

REDUCTION GEARS.

The reduction gears are of the Westinghouse double helical type with hydraulic floating frames, as shown in Figs. 10 and 11. The hydraulic cylinders, which support the floating frame, afford dynamometers for the measurement of power. A Kingsbury segmental thrust bearing is located at the forward end of the gear to take the propeller thrust. At the designed full speed the gears reduce the revolutions from 3,600 to 200, the reduction ratio being 18 to 1. Each pinion transmits 5,500 H.P., the total for one gear being 11,000 H.P.

The gears are completely enclosed in an oil-tight cast-iron box divided on a horizontal plane. The gears are of the double helical type, the teeth being inclined at an angle of 30 degrees. The teeth are of involute form and are cut with the utmost precision to obtain quiet and reliable operation.

The pinion is of the three-bearing type and is carried in a heavy floating frame, which automatically maintains the alignment of the pinion and gear under all conditions of load, thereby producing uniform distribution of pressure over the entire length of tooth face. The pinion is driven by a flexible shaft, which extends through it and is secured to the end distant from the coupling. The flexible coupling permits the pinion to move endwise, and the flexible shaft permits the floating frame to function without restraint. The resulting uniform distribution of tooth pressure at all loads permits the use of higher pressures and smaller pinions and gears than is the practice where rigid bearings are used.

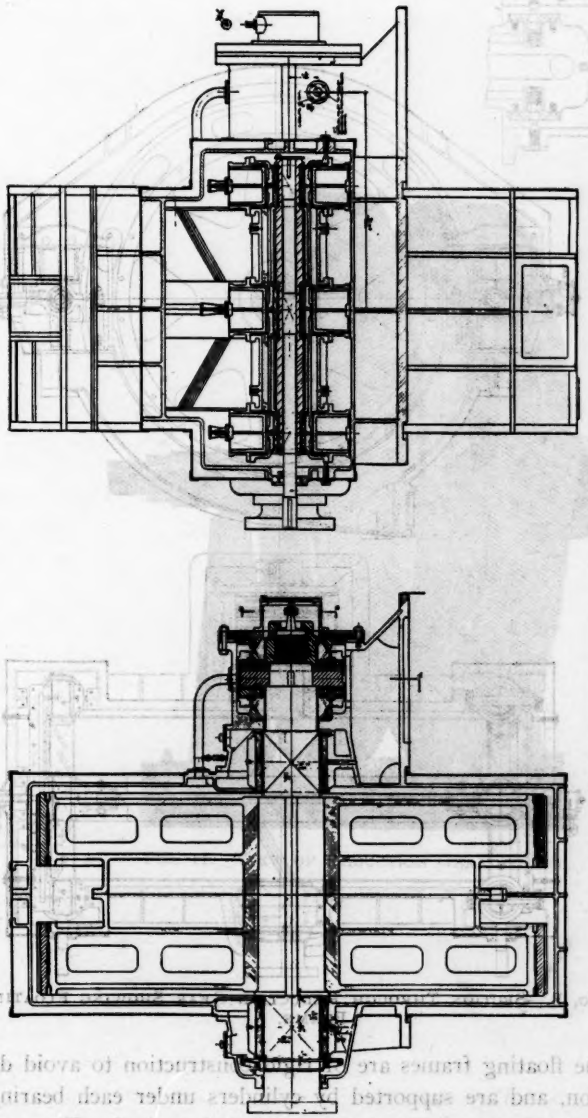


FIG. 10, A.—SECTION THROUGH REDUCTION GEAR SHOWING KINGSBURY THRUST BEARING.

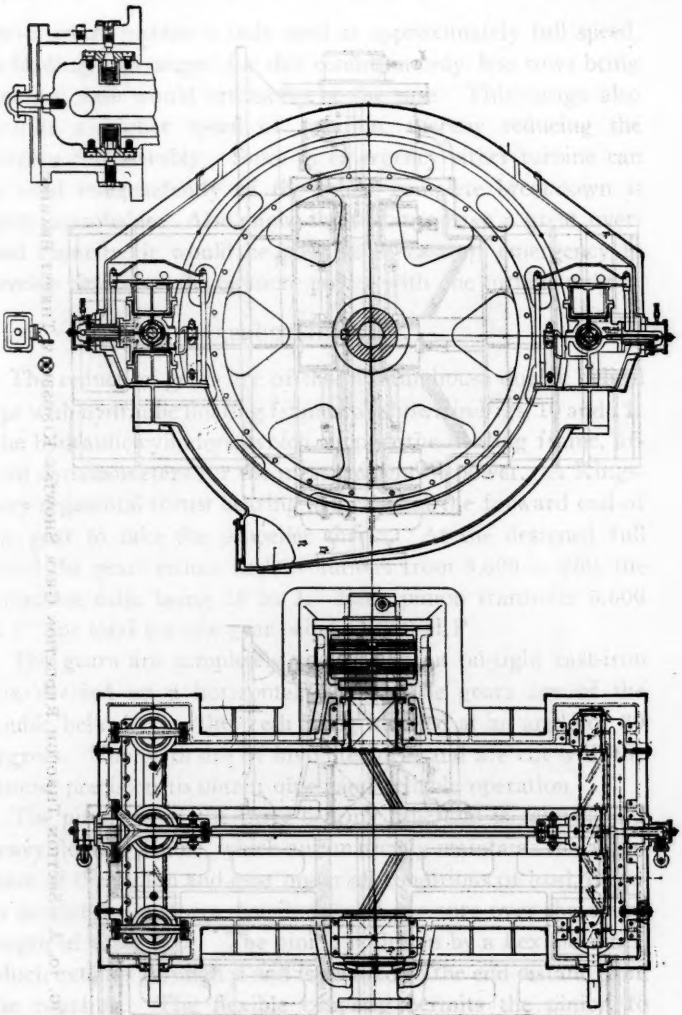


FIG. 10, B.—SECTION THROUGH REDUCTION GEAR SHOWING FLOATING FRAME.

The floating frames are of rigid construction to avoid deflection, and are supported by cylinders under each bearing. The pinions are single forgings of carbon steel. The gear

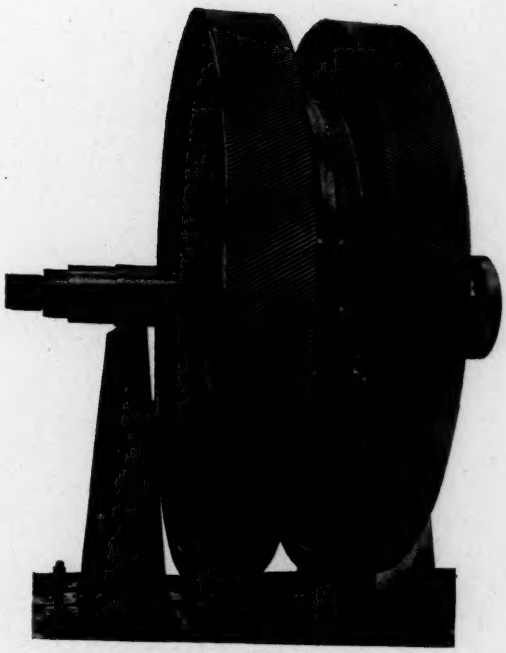


FIG. 11.—VIEW OF REDUCTION GEAR WHEEL.

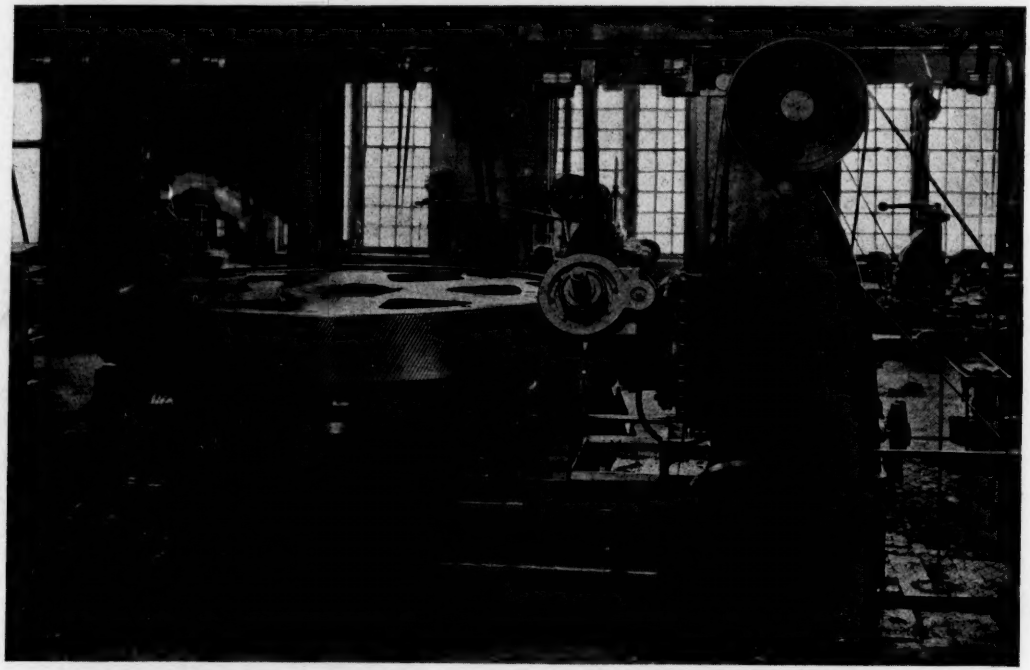


FIG. 12.—VIEW OF GEAR-CUTTING MACHINE IN OPERATION CUTTING A SECTION OF THE GEAR WHEEL SHOWN IN FIG. 12.

wheels are composed of a heavy forged-steel shaft, cast-iron center and rolled-steel rims. The gear and pinion bearings have heavy cast-iron bodies lined with white metal, and are supplied with oil from the main oil system under a pressure of about five pounds per square inch. A copious supply of oil is sprayed on the pinion so as to pass between the meshing teeth and afford the required lubrication and cooling.

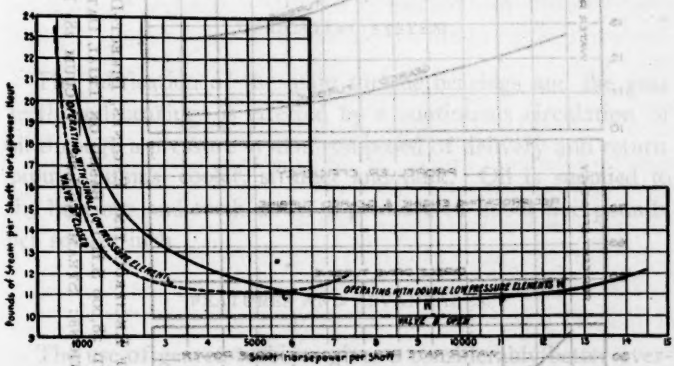

Pounds of Steam per Shaft Horsepower Hour
OPERATING WITH SINGLE LOW PRESSURE ELEMENT IN VALVE CHAMBER
OPERATING WITH DOUBLE LOW PRESSURE ELEMENTS IN VALVE CHAMBER
Shaft Horsepower per Shaft

FIG. 13.—CURVE SHOWING STEAM CONSUMPTION GUARANTEES FOR THE MAIN TURBINES ONLY IN POUNDS OF STEAM PER HOUR PER SHAFT HORSEPOWER AT THE PROPELLER SHAFT UNDER THE FOLLOWING CONDITIONS: STEAM PRESSURE AT TURBINE 260 LBS. GAGE, DRY AND SATURATED. VACUUM IN EXHAUST CHAMBER 28.27 INCHES OF MERCURY (REFERRED TO 30 IN. BAROMETER). S.H.P. FOR ONE UNIT ONLY.

KINGSBURY THRUST BEARING.

The Kingsbury thrust bearing, which takes the thrust of the propeller, is located at the forward end of the gear, and is composed of a single collar, which operates between segmental shoes made of forged steel and lined with babbitt. Each shoe is pivoted on a hardened seat, which permits it to automatically assume the position required to distribute the load uniformly over the entire surface. This uniform distribution of pressure due to the self-alignment of the shoes, gives perfect film lubrication, which permits the use of a considerably higher unit

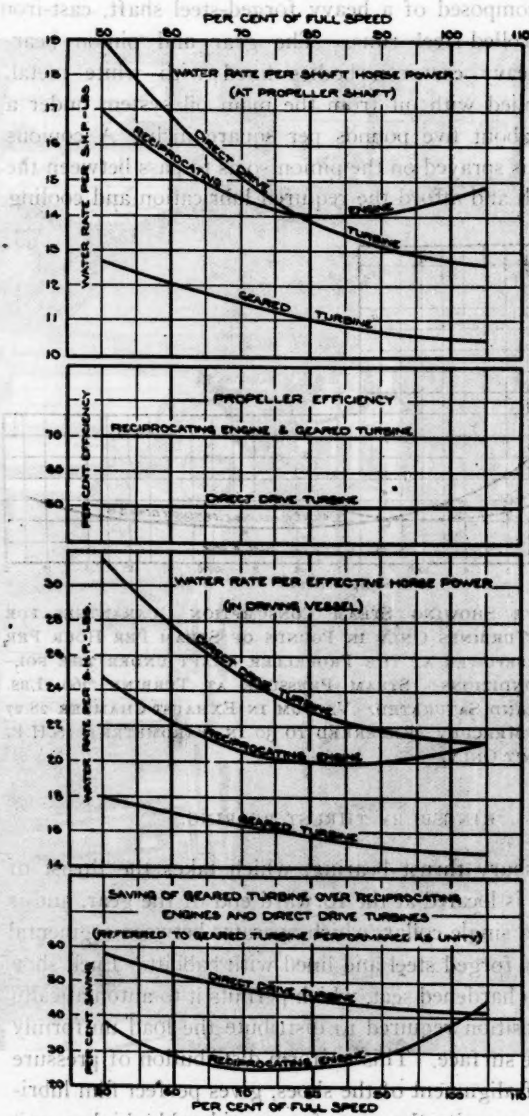


FIG. 14.—COMPARISON BETWEEN PERFORMANCES OF RECIPROCATING ENGINES, DIRECT-DRIVE AND GEARED TURBINES FOR A BATTLESHIP OF 21 KNOTS AND 28,000 S.H.P., BASED ON TRIAL DATA OF RECENT BATTLESHIPS. (OPERATING CONDITIONS: STEAM 250 LBS. VACUUM 28½ IN.)

pressure than is used in ordinary thrust bearings, and also gives greater reliability of operation.

The principal parts of the bearing are made of forged and cast steel. The bearing is lubricated by a continuous circulation of oil, which is admitted near the bottom and completely fills the casing, the outlet pipe being led from the top of the casing into the gear box.

LUBRICATING SYSTEM.

The lubrication of the main turbine bearings and the gear teeth and bearings is effected by a continuous circulation of oil through a pressure system composed of delivery and return piping, pumps, cooler, strainer and tank. Oil is supplied to the bearings and teeth under a pressure of about five pounds per square inch.

FEATURES AND ADVANTAGES.

The use of geared turbines gives a considerably better overall steam consumption than is obtained with existing types of naval machinery. As compared with the direct-connected turbine, it materially increases the efficiency of both the turbines and propellers, thereby greatly increasing the overall efficiency and economy of the propelling machinery. The superior mechanical performance of geared turbine machinery as compared with other types for a 28,000 S.H.P. battleship at 21 knots speed is shown in Fig. 14.

Smaller steam consumption obtained with geared turbines results in:

(*a*) For the same weight of machinery and fuel, greater power, greater speed and greater steaming radius.

(*b*) Or for the same speed and radius of action it enables the designer to give the ship considerably greater offensive and defensive power, due to the saving in weight of machinery and fuel.

The weight of geared turbine machinery is about half of

that required for any other type. In addition to this, of course, there is an even greater saving in boiler weights, fuel, and ship's structure where the same speed and cruising radius are provided. The space occupied by geared turbines is somewhat less than that of either reciprocating engines or direct-connected turbines.

Great endurance and reliability together with practically no repairs result from the use of rotative machinery, which is an important advantage in comparison with reciprocating engines. This feature is even more pronounced with geared turbines than with direct-connected turbines because of the smaller and more rigid structures and rotating parts. As compared with direct-connected turbines, geared turbines have ample backing and maneuvering power. This results from more efficient backing turbines and larger and more efficient propellers, the latter being the same as is used with reciprocating engines.

The divided-flow type of turbine gives a very good economy at reduced power without the extra complication of cruising turbines. The steam consumption at the cruising speed is considerably better than obtained with direct-connected turbines or reciprocating engines. Since naval vessels do most of their steaming at cruising speeds, this is an important item.

In general, the construction of the turbines and gears is heavy and rugged, and ample precautions have been taken to obtain safety and reliability throughout. There are two turbines, on each shaft arranged so either turbine can be cut out and the other operated independently in case of emergency, thus guaranteeing safety against breakdown, which is important in a naval vessel. The vessel could always proceed at nearly the full power in case one turbine should break down. The main turbines are provided with safety overspeed governors, which prevent them from exceeding the maximum safe speed. These governors, however, do not stop the turbines, which would be a disadvantage, particularly in case the machinery was being forced to the limit in a naval action, or in the case of the propellers racing in a heavy seaway.

The turbines have a considerable overload capacity and reversing power, the latter being over 40 per cent. of the ahead full power with the same flow of steam, and considerably more with increased flow of steam during the short reversing period—not over two minutes usually. While the reversing power is less than that of a reciprocating engine, it is quite ample for the requirements.

Many who are not familiar with geared turbines have the impression that with this type of machinery insufficient backing power is characteristic, the same as with direct-drive turbines, from which the present type was evolved. This, however, is not correct. With the direct-drive machinery, due to the unfavorable speed at which they operate, the efficiency of both the turbine and propeller is low, giving a very low overall efficiency. With geared turbine machinery, however, both the turbine and the propeller operate at speeds which give high efficiencies for both, and relatively a much higher overall efficiency or stopping power. It is obvious that by doubling the efficiencies of both turbine and propeller the power of the turbine is doubled, and that of the propeller (the effective stopping power) is quadrupled, which is substantially the relation between direct-drive and geared turbines.

It may be added that the size and efficiency of the propellers are of more importance than high backing power. In stopping the vessel the large propellers used with geared turbines "take hold of the water" better.

The ahead turbines are provided with group control of the nozzles to obtain more economical distribution and use of the steam at all powers.

In addition to the usual steam seals for the turbine glands water seals are also provided for use while the turbines are in normal operation. Since the latter give a perfect seal with absolutely no leakage whatever of steam into the engine room, or air into the turbines, their use is advantageous.

The turbine blading is of the Westinghouse standard type and is particularly rugged and substantial. The blades are

held in the rotor by mechanical interlocks and are amply secured against vibration.

The Kingsbury thrust bearing incorporated in the gear is a great improvement over the usual type of thrust block and its seating.

The small size of the turbines and their more rigid and substantial construction renders the high-speed turbines used in this installation very much less liable to distortion troubles than is the case with the large and comparatively light structures of direct-connected turbines. This is one of the important features of high-speed geared turbines that is often overlooked, but which is most important for naval service where great reliability is essential.

INTERNAL STRESSES IN WROUGHT BRASS AND BRONZE.

By D. J. McAdam, Jr., Associate.

Increased use of wrought brass and bronze in recent years has been accompanied by increased instances of inexplicable failure. These failures occur in metal apparently of excellent physical properties. Sometimes the metal fails after a short time in service under stress apparently well within the elastic limit. Often the material is found broken after an interval of storage.

As in all cases of metal failure, various causes may be assigned. After eliminating all defects of composition and microscopic structure, there remain a large number of failures which cannot be ascribed to these causes. Within the last few years it has been found that a very frequent cause of such failures is high internal stress.

METHOD OF MEASUREMENT OF INITIAL STRESS.

For measurement of these stresses a method has been developed by Professor E. Heyn of Charlottenburg. It may be illustrated by the sketch, Fig. 1.

If a cylinder of length L and radius R is to be investigated, the length is very carefully measured, and thin layers of definite thickness are then turned off, the length of the cylinder being measured after removal of each layer. The original stress in any layer may now be calculated from the total change of length after removal of that layer and of the next preceding layer.

Let l_n and l_{n-1} be the total change of length of the cylinder after removing n and $n-1$ layers respectively; let r_n and r_{n-1} be the corresponding radii; let A_n and A_{n-1} be the correspond-

ing areas of cross section remaining after removing n and $n-1$ layers respectively; let E be the modulus of elasticity. The total force of tension originally exerted by the outer "n" layers must have been equal to the total original force of compression in the inner remaining layers. This total force of compression F may be calculated, using the usual equation for modulus of elasticity: stress = E multiplied by strain, or

$$\frac{F_n}{A_n} = \frac{E l_n}{L}.$$

Similarly the original force of compression F_{n-1} in the inner metal after removal of $n-1$ layers may be represented by the equation

$$\frac{F_{n-1}}{A_{n-1}} = \frac{E\,(l_{n-1})}{L}.$$

The original force in the nth layer is obtained by subtracting F_{n-1} from F_n, a positive remainder indicating tension.

$$F_n - F_{n-1} = \frac{E l_n A_n}{L} - \frac{E\,(l_{n-1})\,(A_{n-1})}{L}$$

$$= \frac{E}{L}\Big[l_n A_n - (l_{n-1})\,(A_{n-1})\Big].$$

The force per unit area, or stress, therefore, in the nth layer was

$$S = \frac{E}{L}\,\frac{l_n A_n - (l_{n-1})\,(A_{n-1})}{A_{n-1} - A_n} = \frac{E}{L}\,\frac{l_n\,(r_n)^2 - (l_{n-1})\,(r_{n-1})^2}{(r_{n-1})^2 - (r_n)^2}.$$

DEFECTIVE MANGANESE-BRONZE PROPELLER SHAFT.

As an illustration of the practical application of this method, the case of a defective manganese-bronze propeller shaft may be described. The propeller shaft was one of several that had been installed in motor boats for the Navy. The shafting had passed inspection and been installed apparently in good condition. After running a comparatively short time, however, numerous cracks developed. The particular shaft

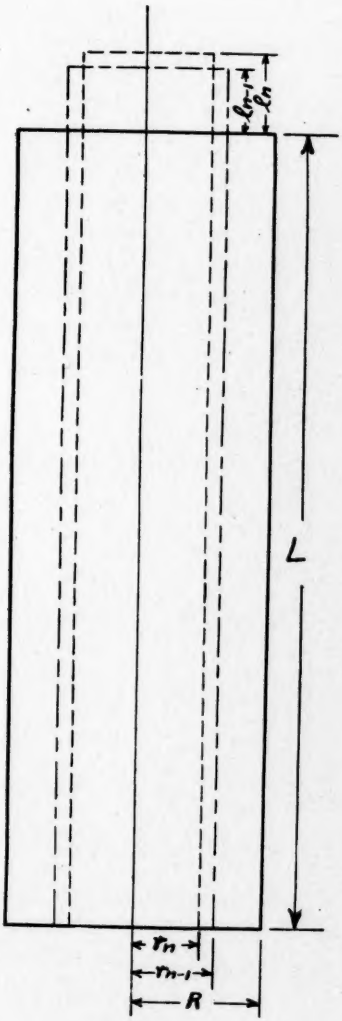


FIG. 1.

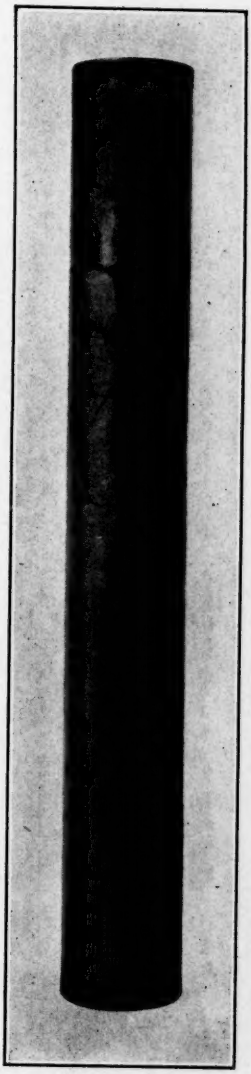


FIG. 2.

FIG. 5.

TABLE II.

Tension Tests.

Specimen number.	Diameter.		Area.			Length between punch marks, inches.	Elongation.		Prop. limit.		Yield point.		Max. stress.		Modulus of elasticity.
	Test piece, inches.	Fracture, inches.	Test piece, square inch.	Fracture, square inch.	Reduction, per cent.		Inches.	Per cent.	Pounds.	Pounds per sq. inch.	Pounds.	Pounds per sq. inch.	Pounds.	Pounds per sq. inch.	.....
17.....	0.505	0.383	0.2	0.115	42.5	2	0.58	29.0	7,600	38,000	10,400	52,000	15,620	78,120	13,850,000
18.....	0.505	0.382	0.2	0.114	43.0	2	0.58	29.0	7,500	37,500	10,500	52,500	15,670	78,370	15,000,000
Average.....	.....	.....	.....	.....	42.8	.....	.....	29.0	.....	37,500	.....	52,300	.....	78,240	.....

examined had been driven by a gasoline engine of about 40 H.P. at about 600 r.p.m. The shaft was about 1¾ inches in diameter. Fig. 2 shows the appearance of the piece of shaft submitted for test.

The chemical composition was as follows:

Copper,	57.0
Zinc,	40.9
Tin,	0.91
Iron,	1.01
Manganese,	trace.
Lead,	trace.
Aluminum,	0.09

TENSION TESTS.

The test pieces were taken from the central part of the shaft after all cracked metal had been turned off. The results are given in Table II. It will be noticed that the elongation of this material is slightly below the minimum (30 per cent.) required by the Navy specifications for rolled manganese-bronze, otherwise the material conforms to specifications.

INTERNAL STRESS MEASUREMENT.

Using the formula developed above, internal stresses were calculated from results of a series of measurements of this specimen of manganese-bronze. The experimental details were as follows: Two marks were made on the circumference of the cylinder at each end and on opposite sides. The length of the cylinder between each pair of marks was then carefully measured by a comparator. Over a definite length L between these marks thin layers were turned off and the length between marks measured after removal of each layer. The portion of the cylinder used for these measurements contained no visible cracks. The temperatures of the metal were carefully observed at the time of measurement, and allowance was made for the effect of temperature change.

The values thus obtained are given in Table I below. In using the formula for stress developed above

$$\left(S = \frac{E}{L} \frac{l_n (r_n)^2 - (l_{n-1})^2 (r_{n-1})^2}{(r_{n-1})^2 - (r_n)^2}\right),$$

the value of E was taken as 14,000,000. This value is about the average of the two obtained by tension tests, and corresponds with the values given in handbooks. In interpreting the stress results given in column 6 of Table I positive values represent tension; negative values represent compression. It will be seen that the results indicate tension in the six outer layers and compression in the other three layers removed. The experiment was stopped at this point because the change in length had nearly ceased, and remaining material was to be used for the above-described tension tests.

TABLE I.—INTERNAL STRESS MEASUREMENTS.

1	2	3	4	5	6	7
Diameter (inches).	Length between marks (inches).		Average total change of length between marks (inches).	Length of reduced section (inches).	Stress in nth layer tension (lbs. per sq. in.).	Layers.
	(*a*)	(*b*)	l_n	L	S	
1.747	6.82219	6.78286	.00074	5.24	29,900	1st.
1.692	6.82270	6.78382	.00282	5.24	87,900	2d.
1.642	6.82495	6.78573	.00416	5.24	48,300	3d
1.592	6.82615	6.78721	.00529	5.24	34,700	4th
1.542	6.82744	6.78817	.00617	5.24	21,100	5th
1.492	6.82814	6.78923	.00805	5.24	17,200	6th
1.392	6.82983	6.79129	.01038	5.24	−4,400	7th
1.192	6.83168	6.79409	.01131	5.24	−22,100	8th
0.992	6.83240	6.79522	.01211	5.24	−25,500	9th
0.792	6.83329	6.79593				

The results recorded in Table I were plotted as shown in Figs. 3 and 4. In Fig. 3 tensile stresses in the various layers are plotted as ordinates, and radii of the layers as abscissas.

In Fig. 4 tensile stresses in the various layers are plotted as ordinates, while sectional areas included within the layers are plotted as abscissas.

SUMMARY OF RESULTS.

The results plotted in Fig. 3 show that the outer layers of this shaft were under great internal tensile stress, while the interior of the shaft was under compressive stress. Near the

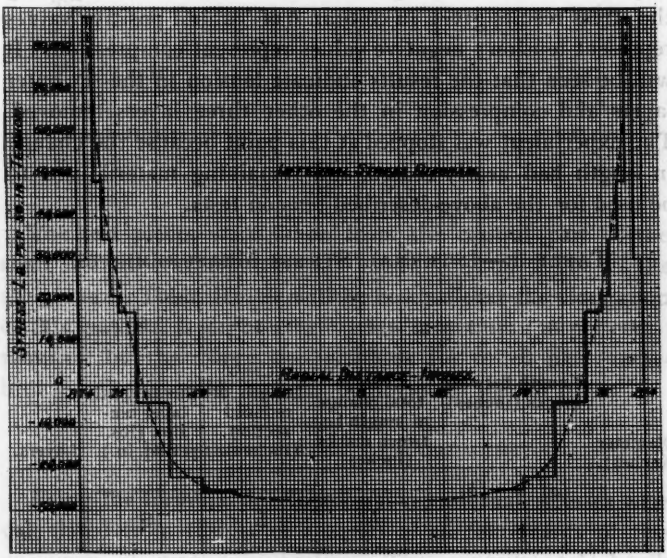


FIG. 3.

circumference of the shaft the initial tensile stress was evidently above the average tensile strength of the metal. In the extreme outer layer, however, the recorded stress is much smaller than in the layer next within. The reason for this abrupt change of stress in the outer layer is discussed below.

The full lines in Fig. 3 show the calculated values as plotted. Since in the calculations the stress was assumed constant throughout each layer, the plotted results form a rectilinear figure. In fact, however, there was undoubtedly

a gradual change of stress from circumference to center of the shaft. This change is illustrated in the figure by the smooth, dotted line curve.

When internal stresses are plotted as ordinates against section areas as abscissas, as illustrated by Fig. 4, the total forces in tension and compression can be calculated and compared. The total forces are represented by the areas included between the curve and the axis of abscissas. Theoretically these total forces in tension and compression must be equal. An actual count of areas in Fig. 4 showed that there are

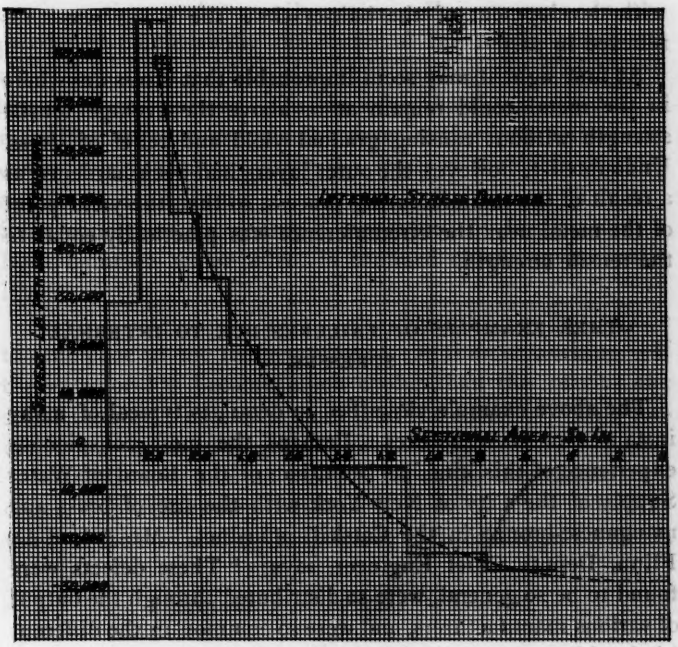


FIG. 4.

approximately 14.4 large squares representing total tensile force and 14.7 squares representing total compressive force. The agreement is, therefore, as good as could be expected.

CAUSE OF FAILURE OF THE SHAFT.

The great internal stress in the outer layers added to the stresses in service undoubtedly started the cracks in the metal. When one considers that the measured internal stress near the circumference of the shaft was considerably greater than the average tensile strength of the metal, and probably was originally as high as 100,000 pounds per square inch at the extreme outer surface, the failure of this shaft in service is easily understood. The lowered stress in the extreme outer layer, mentioned above, was probably due to yield of the metal, with development of the hair-cracks; the stress was thus relieved in part.

At first sight it may seem improbable that stress anywhere in the metal could remain at or above the average tensile strength without causing gradual yield and relief of stress. Undoubtedly, however, the outer layers had been hardened as a result of the manufacturing process, so that the yield point of the metal near the circumference was practically equal to the tensile strength.

OTHER EXAMPLES OF FAILURES DUE TO INTERNAL STRESSES.

The above example, described in detail, is typical of many instances of such failures. Perhaps the most conspicuous experience of this kind has been that of the Board of Water Supply of the City of New York with the use of brass and manganese-bronze in the Catskill aqueduct. Mr. Alfred D. Flinn, Deputy Chief Engineer, says: "These defects were found to be so general, were so distributed through the output of various manufacturers, and were so common in the different kinds of brass, that all wrought brass fell under suspicion."

Fig. 5 shows a piece of cold-rolled manganese-bronze bar kindly sent to the writer by Mr. Flinn. In this bar many transverse cracks are visible. The bar has been slightly bent to make the cracks more conspicuous. Many such specimens

have been tested at the Bureau of Standards and found to have high internal stresses which undoubtedly caused the failure.

CAUSES OF HIGH INTERNAL STRESSES.

The reason for internal stresses in metals may be discussed generally, although the actual method of manufacture in the above described examples is not known. Any process of working metal which causes the center of the mass to be extended more than the outer surface, tends to leave the central portion of the mass under compressive stress and the outer portion under tensile stress. For example, in the process of cold drawing the center of the rod is drawn out more than the circumference, thus causing the above mentioned stress relations. On the other hand, any process of manufacture, that extends the outer surface more than the center, tends to cause compressive stress in the outer portions and tensile stress in the inner portions of the mass of metal. Such conditions, for example, are caused by forging with a light hammer, whose effect is confined chiefly to the outer layers.

Unequal extension of interior and circumference may be due to difference in the temperatures at the time of working. If the exterior has become much cooler than the interior, the softer interior is extended more than the exterior. This condition may occur in forging by press.

In addition to internal stresses set up by mechanical processes, it is well known that such stresses are often caused by unequal rates of cooling in different parts of a mass of metal.

REMEDIES FOR INITIAL STRESSES.

Although, as described above, internal stresses can be regulated somewhat by the process of working the metal, it is usually necessary to employ other means to insure the absence of such excessive stresses. For this purpose, annealing at a carefully regulated temperature and for the proper length of

time is usually most effective. It has been found that dangerous internal stresses can often be relieved in this way without unfavorable effect on the physical properties of the metal. Since, therefore, the prevalence of initial stresses in wrought brass and bronze has been brought to light, there seems to be good reason to hope that manufacturers will take steps to improve the quality of metal in this respect, and that methods of inspection will be modified so as to insure the absence of such dangerous conditions.

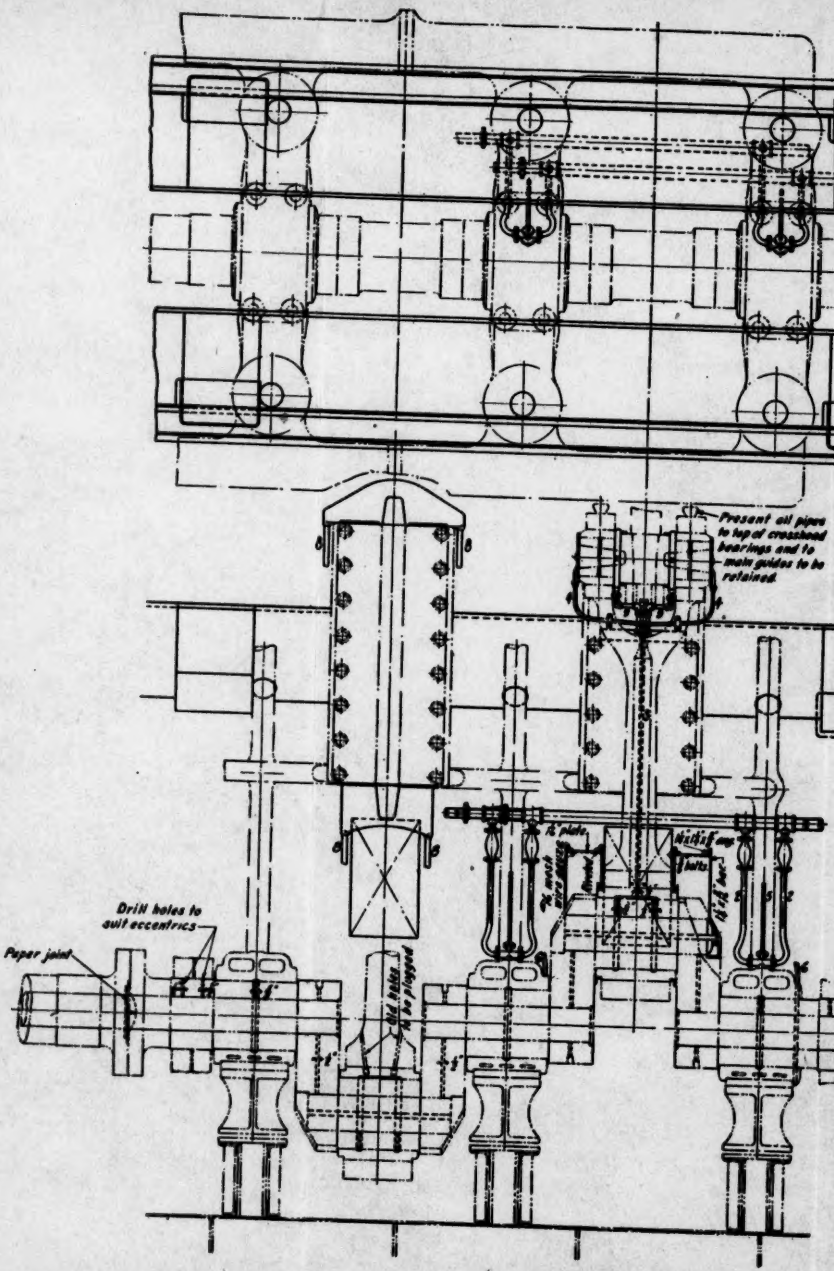
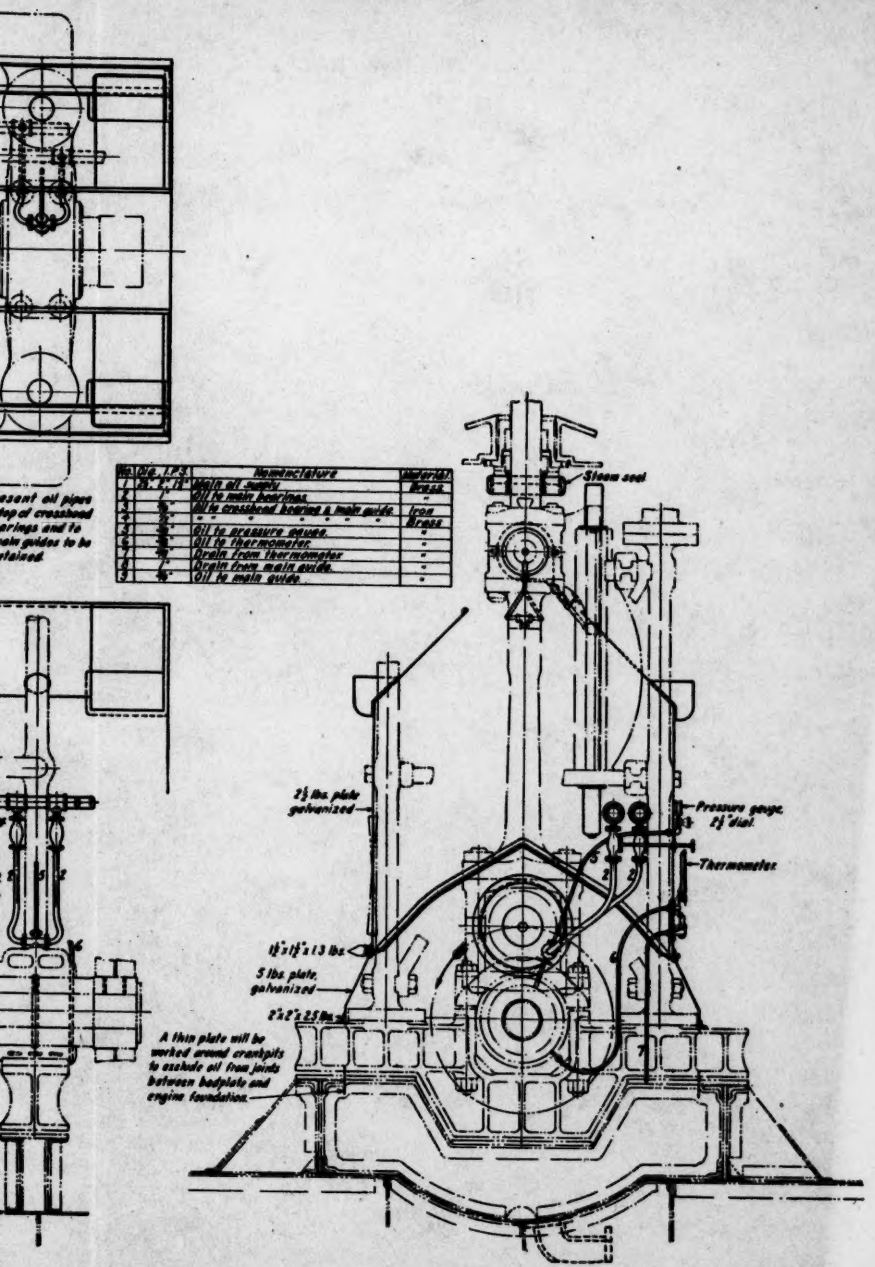

Present oil pipes to top of crosshead bearings and to main guides to be retained
Drill holes to suit eccentrics
Paper joint

FIG. 1.—FORCED LUBRICATION FOR RECIPROCATING


No. Off. I.P.S.
Nomenclature
Material
Steam cock
Pressure gauge 2½" dial
Thermometer
2½ lbs. plate galvanized
5 lbs. plate galvanized
A thin plate will be worked around crankpits to exclude oil from joints between bedplate and engine foundation.

PROCATING ENGINES OF BATTLESHIPS AND CRUISERS.

FORCED LUBRICATION.

By Lieutenant Commander A. T. Church, U. S. N., Member.

The following is a compilation of forced-lubrication data for the most part from reports of officers in the Fleet.

Fig. 1 shows an arrangement of pipes for forced lubrication for reciprocating engines of the U. S. battleships and cruisers.

Fig. 2 shows an arrangement of pumps, tanks, filters, etc., for forced lubrication for reciprocating engines of battleships and cruisers.

Oil is supplied under pressure to main bearings, crank-pin bearings, crosshead pins, crosshead guides, eccentrics, and ahead faces of thrust-bearing horseshoes. At each main bearing a radial hole is drilled though the wall of the crank shaft to the axial hole in the shaft. In the path of this hole an annular recess is cut in the bearing brass. In a similar manner radial holes are drilled through the shaft and each eccentric to the axial-shaft hole. The axial hole in the crank shaft is connected to the axial hole in the crank pin by drilling a hole in the crank web. Radial holes are drilled in each crank pin to the axial hole in the pin. The axial hole in the connecting rod is extended through the crank-pin brass to the crank pin. The result of all these holes is to form a closed circuit for the oil from the main bearing supply pipe to the top of the connecting-rod axial hole. Oiltight caps are fitted at the ends of axial holes in the crank shaft and crank pins. A gage and thermometer is fitted at each bearing.

Referring to figures 1 and 2, in the engine and pump room are located a forced-lubrication service pump, a service and return pump and a return pump; a supply oil tank and a

water and oil-settling tank; three strainers for supply oil tank, two strainers for discharge to engines, one suction-well strainer box and one drain pump. The supply oil tanks are cross connected. Oil is supplied to the bearings by either the service pump or the service and return pump, the oil passing through strainers, either pump being capable of using either strainer and its own discharge line. At each bearing two valves are fitted, one for each oil-feed line, for the purpose of regulating the flow of oil.

The course of oil through the system is as follows: The service or the service and return pump draws the oil from the main supply tank through the suction-well strainer box and discharges through strainers to engine supply pipes. From the main supply pipe branch pipes conduct the oil to the main bearings through holes in the bearing caps. The quantity of oil supplied is regulated by valves at the bearings. Annular grooves distribute the oil through the main bearings and into the crank-shaft axial hole through the radial holes in the journal. The crank pins are lubricated through the hole in the crank web connecting the axial holes in the shaft and web and the radial hole in the crank pin. The eccentrics are lubricated through the radial holes drilled in the shaft and in each eccentric. The crosshead pins are supplied through pipes leading from the connecting rod through the crosshead pin and out through radial holes.

The thrust bearings are supplied by pipes from the main supply line. Valves are fitted at each thrust to regulate the supply. After performing its function, the oil is forced out at the ends of the bearings and drains to the crank pit, which serves the purpose of a trough for catching the oil. From the crank-pit oil well the oil is drawn by the return pump and delivered through strainers to the main supply tank, thus completing the cycle.

The settling tanks provide a means for allowing impurities to settle, after which the cleansed oil can be returned to the system.

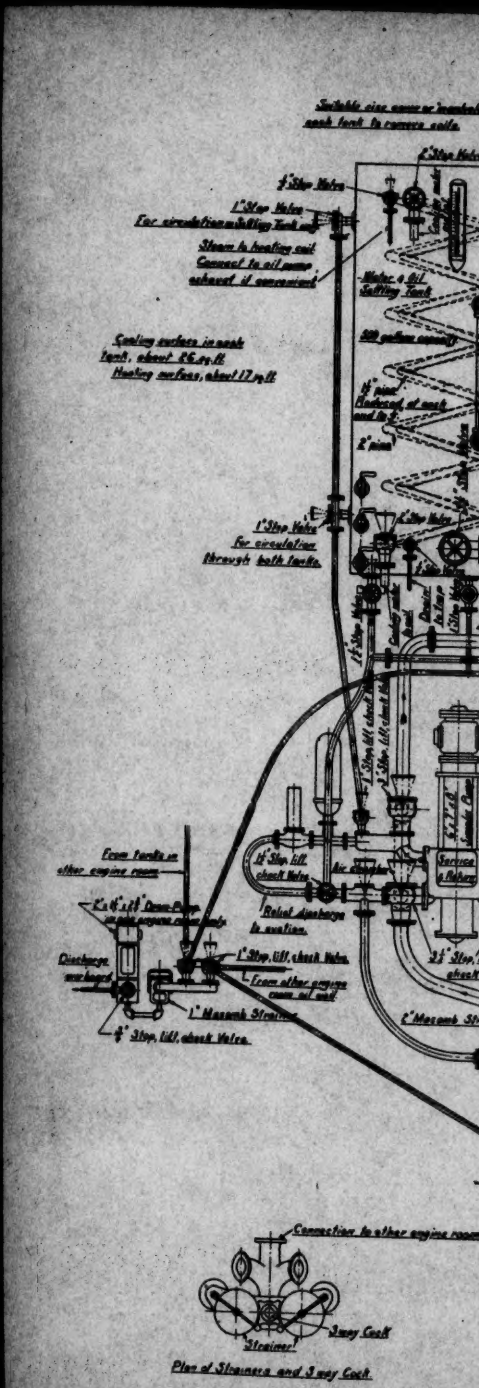
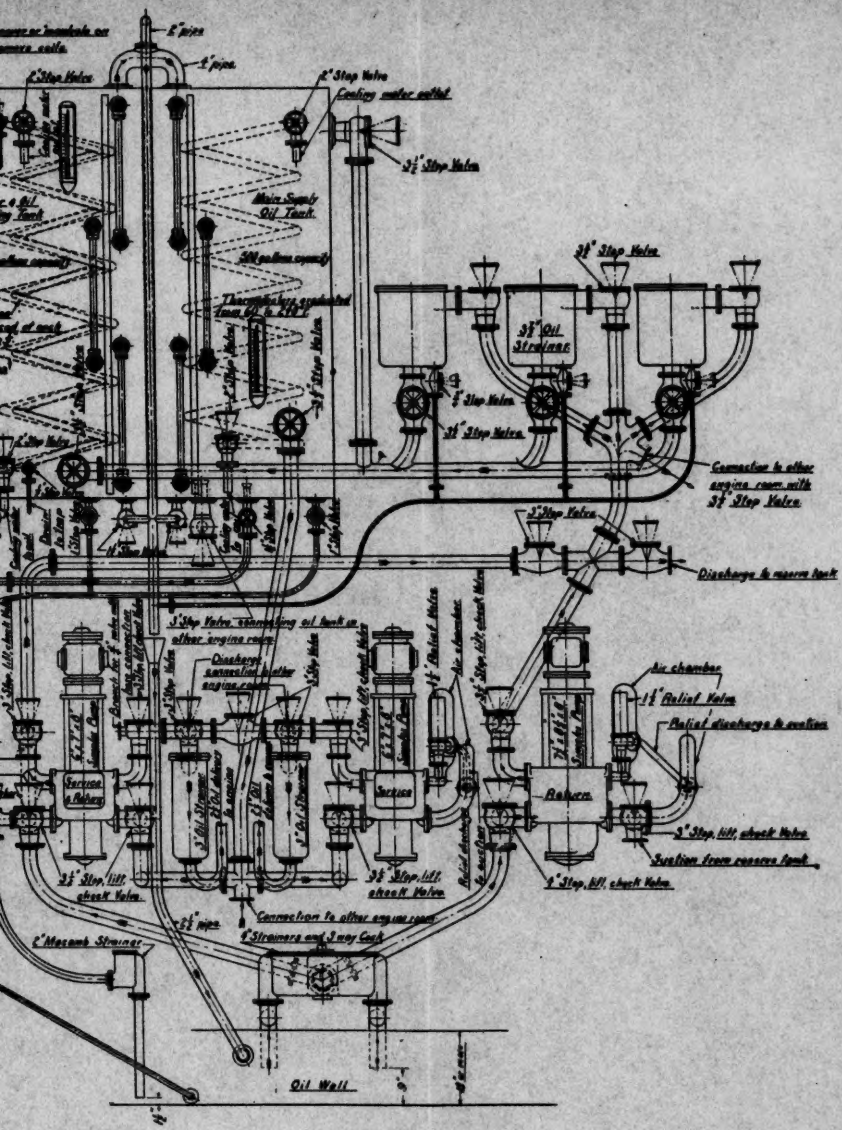

Suitable size cover or manhole each tank to remove coils.
4" Stop Valve
4" Stop Valve
1" Stop Valve
For circulation of Settling Tank oil
Steam to heating coil
Connect to oil pump exhaust if convenient
Water & Oil Settling Tank
200 gallons capacity
Cooling surface in each tank, about 26 sq. ft
Heating surface, about 17 sq ft
4" pipe
Condensed oil each end to E
2" pipe
1" Stop Valve
For circulation through both tanks.
4" Stop lift check Valve
Relief Absorbo in section.
From tanks in other engine room
1" Stop, lift check Valve
Discharge overboard
From other engine room oil tank
1" Mesenk Strainer
4" Stop, lift check Valve
Connection to other engine room
Steamer
3 way Cock
Plan of Strainers and 3 way Cock

FIG. 2.—FORCED LUBRICATION OF



CATION OF RECIPROCATING ENGINES, SHOWING ARRANGEMENT OF PUMPS, TANKS, STRAINERS, ETC.

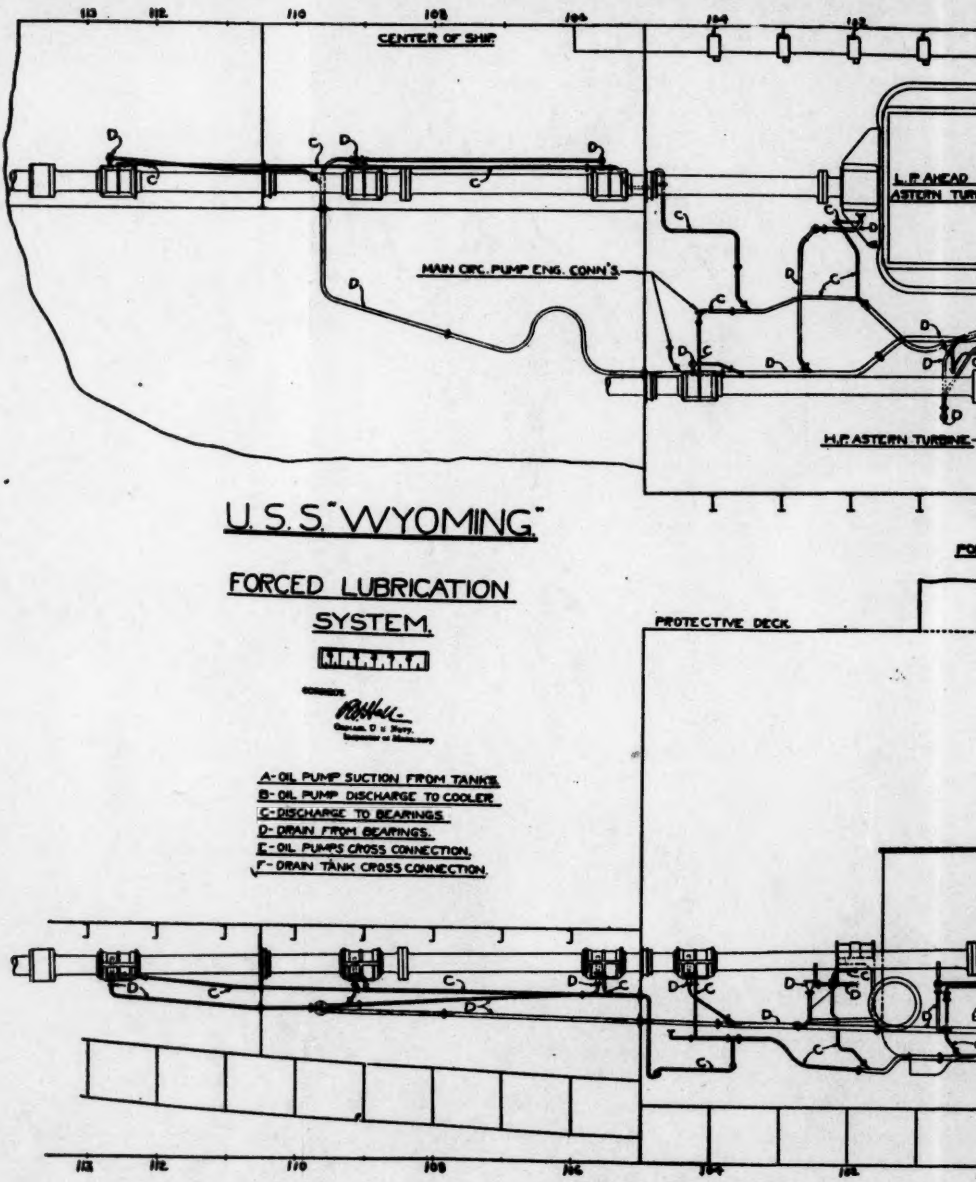
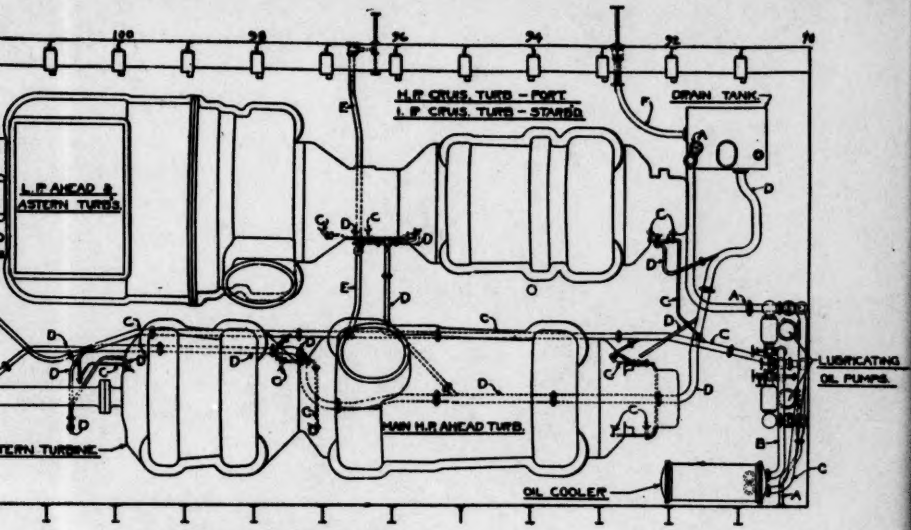
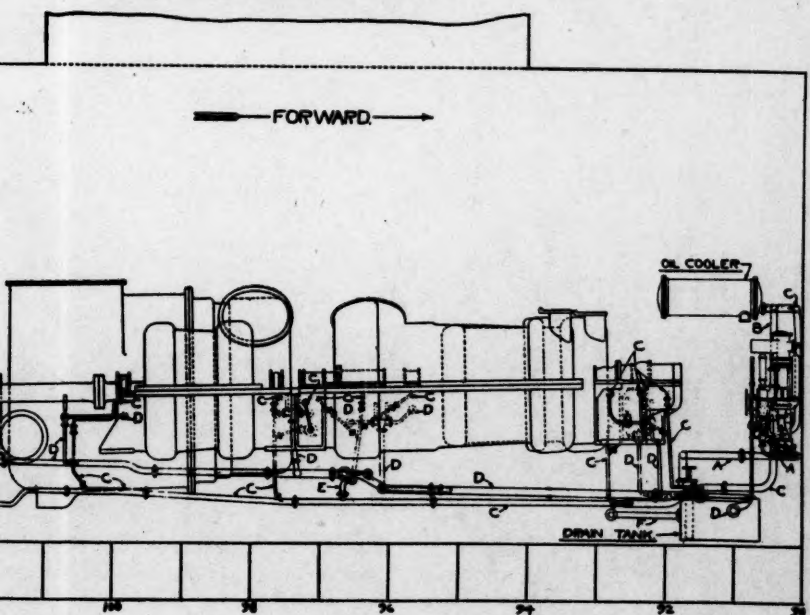

CENTER OF SHIP
MAIN CIRC. PUMP ENG. CONN'S.
U.S.S. "WYOMING."
FORCED LUBRICATION SYSTEM.
A-OIL PUMP SUCTION FROM TANKS.
B-OIL PUMP DISCHARGE TO COOLER.
C-DISCHARGE TO BEARINGS.
D-DRAIN FROM BEARINGS.
E-OIL PUMPS CROSS CONNECTION.
F-DRAIN TANK CROSS CONNECTION.
PROTECTIVE DECK.

FIG. 3.



PLAN OF STARBOARD ENGINE ROOM.
PORT ENGINE ROOM SIMILAR BUT TO OTHER HAND.



G. 3.

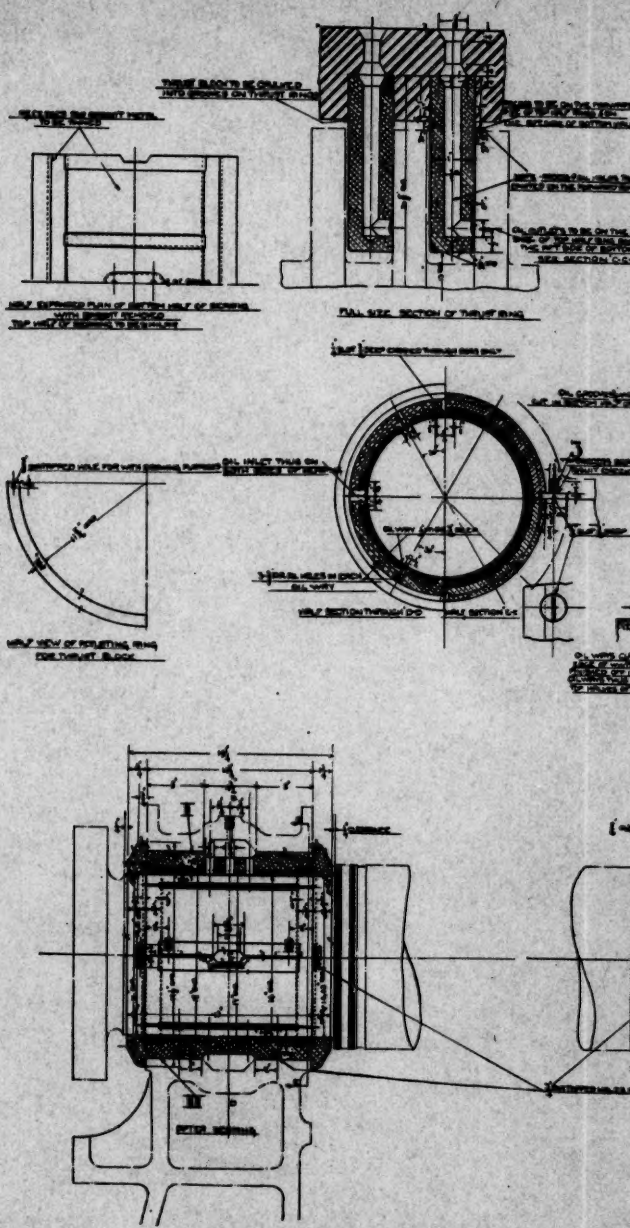
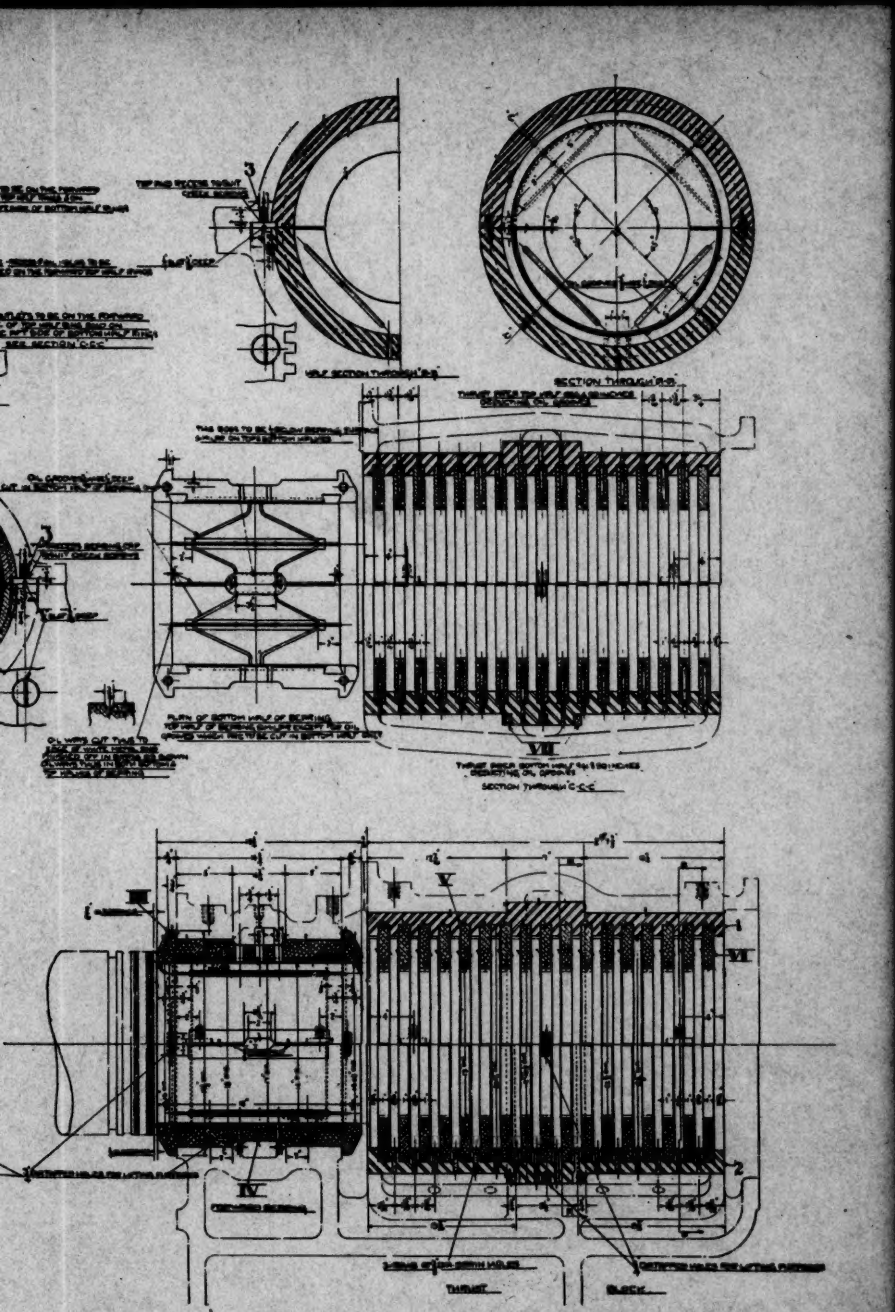


FIG. 4.—DETAILS OF MAIN BEARING AND THRUST



ND THRUST BLOCK FOR MAIN H.P. TURBINE, U. S. S. "WYOMING".

Fig. 3 is a diagrammatic arrangement of the forced lubrication for the turbines of a battleship. The oil pumps take their suction from the drain tank and discharge through a quadruple-flow cooler and thence to the different bearings, from which the oil drains by gravity to the drain tank. Strainers are fitted in the pump suctions. The discharge of oil from each turbine bearing is through a sight-feed glass, so that the flow of oil can be observed. A thermometer is fitted in each discharge. A valve for independent regulation of the flow is placed at each bearing. There is a complete system for the bearings on the port side of the ship and a system similar in all respects for the starboard side, the two systems being cross connected, so that in case of emergency the system on either side will provide lubrication to all bearings on both sides.

Where reduction gearing is fitted oil is led from the main supply pipe to a series of nozzles in the gear case. The nozzles are arranged to spray the oil at the point of mesh in both the ahead and backing directions.

Fig. 4 shows the details of oiling for a main bearing and for a thrust bearing. The oil drains from the bottom of these bearings to the return oil pipe and thence to the drain pipe.

OPERATION.

On reciprocating-engine ships start the pumps from ten minutes to half an hour before getting underway. After starting the pumps they may be run for five minutes to insure that all bearings are getting a plentiful supply of oil and to fill the system so that the working pressures will be attained rapidly when pumps are started, then secure them and start again at the first bell for getting underway. Stop the pumps as soon as word is received from the bridge to secure the main engines, if they have run sufficient time after the last bell for all bearings to receive fresh oil.

On turbine ships oil pumps should be started from two to three hours before getting underway and run after engines are

secured as long as the main circulating pumps are kept running. This is usually from two to three hours, although the *Florida* runs them from six to eight hours.

PRESSURES.

Pressures used by different vessels are as follows:

U. S. S. Texas (Reciprocating Engines): On the main engines the pressure usually carried at the pump is 20 pounds. Running at full power this pressure is increased to 30 pounds.

U. S. S. New York (Reciprocating Engines): Pressures carried on discharge side of strainers are from 20 to 30 pounds gage.

U. S. S. New Hampshire (Reciprocating Engines): Between 10 and 20 pounds by the oil-pressure gages on the various parts of the system.

U. S. S. Delaware (Reciprocating Engines): Fifteen to 35 pounds, depending upon viscosity of oil and speed of main engines.

U. S. S. Michigan (Reciprocating Engines): An oil pressure of 10 to 25 pounds is carried, depending upon the speed of the engines.

U. S. S. Florida (Parson Turbines): The pressures carried on Parson four-shaft turbines are 50 pounds at pump and 10 to 12 pounds on farthest bearings.

U. S. S. Arkansas (Parson Turbines): For cruising speeds the starboard oil-pump discharge pressure is 50 pounds gage, the port 65 pounds gage. This gives an average oil pressure at the bearing of about 5 pounds gage on each engine.

U. S. S. Utah (Parson Turbines): Pressure at pump before passing through cooler, 25 pounds; pressure in system after leaving cooler, 12 pounds; average pressure on bearings, 9 pounds.

U. S. S. Wyoming (Parson Turbines): Pressure carried at farthest bearings for 10 knots and above, 10 pounds, for below 10 knots, 9 pounds.

LEAKS.

There are two classes of leaks: (a) oil, (b) water. Either class may be detected by the daily record of the quantity of oil in the drain tanks. If this tends to decrease abnormally it indicates an oil leak in the system, which is easily detected by inspection of bearings and piping. If, on the other hand, the drain tank gains oil it is an indication that there is a leak in the oil coolers permitting the circulating water to flow into the oil. The leak itself is detected by a pressure test of the cooler. No difficulty should be encountered from leaks when reasonable oil pressures are carried.

Oil leaks in coolers may be detected by means of a small pipe tapped into the highest point of the salt-water side of cooler. This pipe serves as a means of getting rid of air, and will show oil if any is leaking from the oil to the water side through the tubes or ferrules of the cooler.

If the oil gages show a sudden drop in pressure and the supply pump speeds up, inspection should be made to find the cause, which is usually the flooding of some part by opening the feeder valve too wide. If the pressure gradually drops and the supply pump speeds up accordingly, the trouble is usually due to wear of grasshopper joints to crosshead bearings or in main bearings, and should be remedied in port.

If the baffles at the ends of bearings are injured, oil is thrown out as the shaft revolves, and may be thrown out if oil pressure on the bearing is too high.

A ready means of remedying leaks in engine casings is by filling cracks with litharge and glycerine.

UNDERWAY.

A careful watch should be kept on the pumps to see that they do not lose their suction and to keep the pressure constant. A careful watch should be kept on the temperaure of the main bearings and thrust bearings, and the system should be frequently examined for the purpose of detecting leaks and to see that the oil is running freely.

Oil strainers should be cleaned every watch.

Effort should be made to keep the temperature of oil leaving the coolers not less than 90 degrees nor more than 110 degrees.

A record of average bearing temperatures under various conditions of speed, temperature of circulating water, etc., is of considerable value in determining whether or not temperatures are normal under varying conditions.

TANKS, DRAIN POCKETS.

Where accessible, the drain pockets under main bearings should be frequently opened, drained and cleaned, as a very heavy sludge is apt to collect there.

Frequent inspection and care of the strainers in the system will aid in keeping the tanks in good condition.

Tanks should be inspected as a matter of monthly routine. When tanks are inspected oil should be pumped to a treating tank, when available, and the oil treated by raising the temperature high enough to precipitate all water and sediment. The tanks themselves should be boiled out and swabbed.

It is not necessary to allow the oil to cool after heating before drawing off the water. The settling out is practically all done while the mixture is hot, and waiting for it to cool has little or no effect.

OILS.

Conditions in forced-lubrication systems are quite different from those in sight or wick-feed systems. Since the oil is used over and over again, any that may be squeezed out from between the bearing surfaces does not represent a loss; also the oil can be supplied in any amount, thereby eliminating the possibility of an insufficient supply to the bearings. Therefore a lighter oil may be used than with the wick-feed.

Every effort should be made to exclude water, since the oil must be kept in continuous circulation, and the effect of a water leak soon becomes cumulative. One of the primary requisites of a forced-feed oil is that it must not saponify and must

separate readily from water. Too much care cannot be exercised to keep the oil clean and free from the least grit that excessive wear of bearings may be avoided.

TESTS.

Heat sample up to 400 degrees F.; on cooling, oil should not be much darker than it was before heating.

Fill a bottle one-quarter full of oil and then add twice as much boiling water as there is oil, shake well, and allow mixture to stand or settle. There should be no deposit, and the line of separation between the oil and water should show practically no paraffine or solid matter. The oil and water should separate wholly, and there should be no emulsification or saponification.

Fill a test tube with oil and strained gasoline. Shake well and allow to stand for a few minutes. The gasoline, oil and any water that may be in the original oil will form three distinct layers, and the test gives an indication as to whether or not the oil requires heat treatment for elimination of water.

The Bureau of Steam Engineering requires that forced-lubrication oil for main engines must be a pure mineral oil with sufficient body, must not form an objectionable emulsion when steam is bubbled through it; and when stirred vigorously for five minutes with cold or boiling distilled water or salt water, must settle out promptly, leaving as a deposit only a small percentage of separated matter.

Suitable oils for forced-feed lubrication are Texaco Ursa (heavy oil; 750 viscosity at 100 degrees F.); Texaco Algol (heavy oil; 500 viscosity at 100 degrees F.); Texaco Alcaid (medium oil; 300 viscosity at 100 degrees F.); D. T. E. heavy (medium oil; 300 viscosity at 100 degrees F.); Texaco Cetus (light oil; 200 viscosity at 100 degrees F.).

Engineer officers should acquaint themselves with the Bureau of Supplies and Accounts' memorandum, which shows oils suitable for different purposes, so that the wrong oil will

not be used when a special kind of oil is necessary. In some cases a very high-priced oil has been used when a much cheaper oil would answer satisfactorily, and in other cases complaints have been received when a low-priced oil has been used for the wrong purpose. The lubricating value is dependent upon the quality of the oils, and a mixture of different oils gives a result closely approximating the average lubricating value of the oils mixed. In forced-lubrication systems it is permissible to add oil of a different brand to make up losses, and oils may be changed without it being necessary to take all of the old oil out of the system before the new brand can be used.

BOILER GASKETS AND PACKINGS FOR THE NAVY.

By Lieutenant Commander H. T. Winston, U. S. N., Member.

(1) General Remarks.
(2) Physical Properties of Packing and Gasket Materials.
(3) Packings Used for the Wrong Purpose.
(4) Purchase and Choice of Packings.
(5) Simple Tests for Packings.
(6) Standardization Desirable.
(7) Advantages of Annual Contracts.
(8) Packing Contractors.
(9) Types of Navy Contract Packings.
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(12) Types of Boiler Gaskets.
(13) Simple Tests for Boiler Gaskets.
(14) Effects of Boiler Compound, Water and Heat.
(15) Using Gaskets.
Appendix:
Tests of Packings.
Tests of Boiler Gaskets.
Navy Department Specifications.

(1) GENERAL REMARKS.

As any one handling machinery has no doubt on various occasions experienced trouble with gaskets and packings, and the Navy is a great public institution which has worked on the problems involved for many years, some of the results of these investigations, tests, and the consequent conclusions arrived at, should be of interest to the engineering profession.

The types of packings and gaskets actually being used and found most desirable will be shown later. Also, in appendices will be found specifications for a number of types, and information with regard to tests required at the Engineering Experiment Station, Annapolis, Md., for such kinds as cannot be well covered by ordinary specifications, but are purchased as a result of meeting test requirements. In effect these special tests serve to establish "acceptable lists" of certain types of packings and gaskets which have met the severe test require-

ments. It is necessary to know something of the components of packings and gaskets to be able to efficiently select and use them.

(2) PHYSICAL PROPERTIES OF PACKING AND GASKET MATERIALS.

Rubber compositions, containing perhaps only a small amount of rubber, are generally needed to resist the leakage of water and to provide elasticity or resiliency. They are, however, especially the poorer grades made of shoddy or old rubber, subject to rapid deterioration, drying out, cracking and becoming brittle. Heat burns or oxidizes rubber, slowly or rapidly, depending upon the amount of heat applied, as any one knows who has used an ancient pair of overshoes or noticed an old rubber band around a package of papers. Gasoline, oil and other substances readily attack and dissolve or "eat up" rubber. To sum up, rubber is an expensive article and has ordinarily a short life of usefulness.

Rubber cement or compound ("friction") is used to hold the different parts of built-up gaskets or packings together, and to impregnate or coat asbestos or other fiber in such a manner that it will not become water-soaked and fall to pieces. The quality of this compound is most important in spiral-rod packings or boiler gaskets made of woven asbestos cloth, and in compressed sheets of asbestos fiber. The use of an unsuitable compound is a very common fault of manufacturers, with the result that many packings and gaskets soon go to pieces in service, and some cannot even stand being kept on hand a few months in the ordinary storeroom.

Some of the compounds which have given best results contain barium sulphate, zinc oxide, about 15 per cent. of rubber and 2 or 3 per cent. of sulphur. Those which look like white lead, or have a sort of chalky appearance and can be easily flaked off, may always be counted upon to prove inefficient in service.

Asbestos fiber is essential where high heat must be resisted

and metal alone cannot be used. Only the long-fiber asbestos, which is hard to obtain, will weave into a proper cloth or other structure necessary to make a high-grade boiler gasket, or asbestos rope or spiral-rod packing. The shorter fiber asbestos can be spun and woven with the addition of considerable cotton, but heat will soon burn up the latter.

As asbestos readily becomes water-soaked it must be protected from water by a protective coating or cement, which latter is liable to deterioration if exposed to oil or gasoline.

Pure asbestos alone does not hold together well, and in the loose form will soon be washed away by water, or fall apart when subject to vibration unless placed in a bag of asbestos cloth or other container.

Short asbestos fiber is much cheaper than the long fiber, and entirely suitable for use when compressed and firmly cemented together to form a lagging material or sheet packing, except that very thin sheets require somewhat longer fiber than thick ones. It is not generally known that each separate fiber of asbestos, as mined, is in the form of a complete cylinder.

Wire insertion, brass-wire mesh, or, under present conditions, copper-wire mesh as a substitute, is used to strengthen rubber composition sheets. Also extra-strength asbestos cloth is made from strands of asbestos spun around wire. This type is not suitable for piston-rod packings, since the strands of wire soon tend to score the moving rod.

Metals used in making gaskets or packings should be soft and compressible, such as copper, brass, monel metal, white metal, etc. They must be non-corrosive under the conditions to which they are subjected in service or coated to protect them from liquids causing corrosion, and when subject to vibration or heat must not melt nor soon become brittle. When metal is used to form a rigid-type gasket, it is necessary that the gasket be carefully made the exact shape and size required. Corrugated copper gaskets have given fairly good results in making tight joints, but their general use is not desirable. A great many different sizes have to be carried as spares, they

are not easily cut from the copper sheet, vibration and heat are likely to make a gasket in use brittle, and the use over again of an old gasket requires first the trouble of annealing and second reforming in a special corrugating machine. With the softer composition sheet packing any sized gasket is easily cut and the old gasket quickly renewed.

A number of *braided metallic packings and packings reenforced with strips of metal,* intended for use in slip joints and other steam stuffing boxes, are on the market. They are expensive and soon form under heat in a stuffing box a hard, solid mass which is difficult to remove. Where movement of a piston rod or gland is experienced a little wear is likely on the surface of the solid mass of metal, causing a steam leak. Such a leak cannot be stopped by setting up on the gland. When there is no motion very good results may be obtained with braided metal packings. In a few cases they have lasted for years, but their general use is not desirable. Many soft metal composition packings in the form of shot, small bars, etc., are largely advertised. The writer considers the regular "soft packings" far more desirable for general use.

A *good composition for small metallic packing rings* is 88 per cent. lead, 8 per cent. antimony and 4 per cent. copper. Navy antifriction metal, or babbitt (about 4 per cent. copper, 89 per cent. tin and 7 per cent. antimony) may be used as a water packing, but fuses too easily for use with steam.

The *best types of built-up sectional metallic packings must be used for main engine piston rods,* and in other places where a moving rod has side play. A composition of 38 per cent. copper and 62 per cent. lead has given good results.

Flax is required for gland water packing where strength and lasting qualities are needed. Long fiber Russian flax, which combs out to give a brush at least a foot long, is the best and is desirable for stern-tube glands or pump plungers. Italian flax gives fairly good results. Substitutes of jute, hemp, etc., easily become water-soaked and soon go to pieces, especially when around a moving rod or shaft.

Tallow, greases and oils are used as a protective coating to

prevent absorption of water and as a lubricant. They should not be of such a nature that they become rancid or form acids. When a moderate amount of heat is to be encountered tallow or a heavy grease with a high melting point is necessary. Light oils and greases are likely to melt and run out of or be squeezed out of packings. Gland packings saturated with light grease are simply a waste of good material, since nearly all the grease is squeezed out when a stuffing box is set up tight; heat alone or heat and pressure soon make the lubricant a matter of ancient history.*

Cotton is used in the form of cloth to strengthen composition or built-up packings, to provide the necessary fiber for spinning asbestos into yarns, and as a pure cotton cloth or yarn; for example, corset lacing. It burns easily and will not stand high heat; water is readily absorbed and will rot the fibers unless some manner of water-proofing is resorted to.

Compressed wood fiber, or other vegetable fiber, impregnated with a paste of sodium silicate, is used as a packing for gasoline and oil joints. A common form of gasket of this type is one of tough paper, which often gives excellent service under many different conditions provided joint surfaces are smooth enough to permit the use of such a thin gasket.

Graphite is used as a lubricant on the surface of packings and has the advantage of resisting high heat. In some cases gland packings are heavily coated with graphite, most of which falls off in handling and is wasted. Attempts have been made to utilize graphite as a core for gland packings, with the idea that it would gradually work out to the surface and lubricate it. Such attempts have not been successful, and, in fact, even in the best packings, only a small portion of the graphite, which forms a smooth glazed outside surface, appears to serve any useful purpose.

Graphite on one side of a gasket prevents that side sticking when the joint is broken again.

Red lead on the side of a gasket hardens and firmly holds

* For further information about oils, greases and graphites, see an article on "Navy Lubricants and Lubrication" in the May, 1917, JOURNAL.

that side of the gasket to the flanged surface. It also tends to fill up cracks or inequalities of the surface.

(3) PACKINGS USED FOR THE WRONG PURPOSE.

Careful regard to and understanding of the ingredients which make up a packing will prevent the use of a perfectly good packing where it is wholly unsuitable. Often the packing is blamed when the ignorance of the user causes the trouble.

Common mistakes are to expose rubber to heat, gasoline or oil; to cut up a high-grade packing with a rough pump or piston rod; and to use a low-grade valve-stem packing where a high-grade spiral-rod packing is needed.

(4) PURCHASE AND CHOICE OF PACKINGS.

Most manufacturers claim to have secret processes of manufacture and special ingredients in their packings which will cause unusually good results. Those who offer something claimed to be far superior to anything else on the market, or said to give wonderful results under any and all conditions, seldom can supply an article as good as the ordinary standard brands. It may be stated that packings are in a class similar to patent medicines—both are often claimed to be "cure-alls," the public knows little about what they contain or should be composed of, and extravagant advertising or enterprising sales agents may produce a wide sale of inferior products. When possible a reliable company should be dealt with, and efficiency in actual service guaranteed in a written contract, which states that all packings found unsuitable are returnable as rejected material without any expense whatever to the purchaser.

As a rule, special and unusual patented packings are to be avoided. Generally only the manufacturers know how to install and use them properly, they have to be specially made up to fixed dimensions and ordered in advance of requirements from the particular firm which makes them, it is probably

necessary to carry a large number of spares for emergency use, and the cost is nearly always much greater than for ordinary types.

(5) SIMPLE TESTS FOR PACKINGS.

Any type of packing which shows marked deterioration when kept in a storeroom, office or elsewhere for a few months under ordinary conditions, is evidently unsuitable for general use. Some rubber composition sheets dry out and become brittle, so they crack easily when bent with the hands; the cement, used to hold together the layers of composition sheets, "loses its life" and the layers come apart; certain piston-rod, or other spiral-gland packings, get into such shape that they almost fall to pieces, or loose graphite falls off, or light oils melt, ooze out and evaporate. A simple test as to whether or not a packing is subject to rapid deterioration in store, is to bake it for a couple of days in a dry heat of about 212 degrees F. The good packings will not show much change.

A packing supposed to stand a moderate heat should not contain much rubber or burn easily. If a strip of it can be readily lighted with a match and burned, it is not suitable. Often good information as to a packing's composition is obtained by soaking a small piece in gasoline over night. This softens and dissolves the rubber cement, enabling an examination of the other substances in the packing, and shows how it will stand up when attacked by gasoline or oil.

Asbestos packings intended to stand high heat may be burned in a hot flame. They should not burn when gentle heat is applied, and a hot flame should leave the general structure intact. A large cotton content would burn out, leaving the burned mass in such shape that it easily crumbles to pieces.

Packings intended for steam and hot-water joints should effectively withstand four or five hours' boiling in water, or exposure to live steam at 200 pounds' pressure for four hours, without material change.

High-grade asbestos cloth should be a closely woven cloth

of long fiber asbestos, showing not less than 20 strands of two-ply yarns on one cut edge of the cloth and not less than 11 strands on a cut edge at right angles to the first. By pulling apart a yarn the asbestos fibers may be seen. They should be at least an inch long; also when a yarn is burned with a match it should not light very easily, particularly if the "friction" has first been washed out with gasoline, and the yarn should remain practically intact after burning.

High-pressure steam spiral-gland packings for steam-line slip joints, pump-rod stuffing boxes, etc., must have a firm and compact structure. A large rubber core, which quickly hardens under heat, is objectionable; a creamy or yellowish colored cement or friction between the layers of cloth seems best, and the body of the packing should be so built up that it will not fall to pieces or become water-soaked after a little wear.

Simple tests and examinations cannot, of course, replace the elaborate laboratory and service tests, but experience has shown that any one can, with little difficulty, by means of such simple tests, reject the majority of unsuitable packings.

(6) STANDARDIZATION DESIRABLE.

Simplification and standardization is an axiom when efficiency is desired. In a large organization like the Navy, with its personnel ashore and afloat in all parts of the world, originally drawn from many localities, and constantly changing and being transferred from one place to another, standardized methods and material are particularly important. Hence, as few types of packings and gaskets as possible are chosen and supplied as standards to the whole establishment. These are shown a little later, and are such that all are obtainable from a good many different manufacturers.

(7) ADVANTAGES OF ANNUAL CONTRACTS.

Standardization is obtainable only when large amounts of materials are advertised for and purchased through a central agency. This agency must also supervise the drawing up of

the specifications for the materials, the tests conducted, instructions sent out with regard to issue and use of contract supplies, their inspection, and, finally, must collect and tabulate data concerning results obtained in actual service. The latter may prove difficult when little information or inaccurate reports comprise much of the data received.

After receiving the technical advice and recommendations of the Bureau of Steam Engineering, the Bureau of Supplies and Accounts of the Navy Department draws up the contracts and arranges for the distribution of the contract materials, through Paymasters (Supply Officers in this case), to the various vessels and shore stations. Other bureaus of the Navy Department generally also use these packings and gaskets, but may purchase special types if they so desire.

Standard specifications, standard types of packings, lists covering brands tested and found acceptable, and annual contracts covering the standard types, do away with small local purchases at high prices of possibly inferior materials which would involve considerable labor on the part of many individuals, and supply everyone who has had a little mechanical experience with something he can use intelligently.*

(8) PACKING CONTRACTORS.

Some of the *firms who have bid for and obtained packing or gasket contracts with the Navy in recent years are*: (The names of all appear in the "Government Advertiser"): B. F. Goodrich Co., Peerless Rubber Mfg. Co., Crandall Packing Co., Manhattan Rubber Mfg. Co., Revere Rubber Co., Consolidated Packing & Supply Co., Durabla Mfg. Co., Johns-Manville Co., Goodyear Tire & Rubber Co., Montgomery Bros., Inc., U. S. Rubber Co., Anchor Packing Co., New York Belting & Packing Co., Belmont Packing & Rubber Co., Flexitallic Gasket Co., Danubil Co., Gutta Percha & Rubber Mfg. Co., Fibre Finishing Co., Inc., Keasby & Mattison Co., Asbestos Textile Co., Inc., Philip Carey Co., Raybestos Co., and Vulcan Asbestos Mfg. Co.

* Further information about annual contracts will be found under "Navy Lubricants and Lubrication" in the May, 1917, JOURNAL.

(9) TYPES OF NAVY CONTRACT PACKINGS.

† Test required before being considered acceptable. Not covered by standard leaflet specifications.

Kind.	Purpose.	Cost of 1/16 sheet per sq. yd.
†(*a*) *Superheat sheet* of compressed asbestos fiber.	High-pressure steam or internal-combustion engine joints	$1.70
†(*b*) *Asbestos cloth sheet.*	Rough-joint surfaces where (*a*) cannot be used	3.00
(*c*) *Asbestos cloth tape.*	For use where more economical than (*b*)	4.00
†(*d*) *Rubber sheet, wire insertion.*	For use in special cases (hot water and exhaust steam) instead of (*a*) or (*b*)	3.30
(*e*) *Rubber sheet, cloth insertion, Grade No. 1.*	Permanent cold-water joints, not expected to be broken	2.60
(*f*) *Rubber sheet, cloth insertion, Grade No. 2.*	General use for cold-water gaskets	0.90
(*g*) *Compressed wood-fiber sheet,* or tough paper.	Gasoline, oil and water joints	1.15

Kind.	Purpose.	Cost of 3 ft., ½" diameter.
†(*h*) *H. P. piston-rod packing,* spiral form of woven asbestos.	Stuffing boxes of rods of engines and pumps, and steam-line slip joints	$0.60
†(*i*) *Valve-stem packing,* impregnated asbestos rope or cord.	Steam glands of globe-valve spindles, etc	0.40
†(*j*) *Flax packing,* braided flax, soaked in tallow or other grease.	Cold-water packing for stern glands, pump rods, pump plungers, etc	0.25
(*k*) *Tucks packing,* square or round, layers of canvas and hard rubber.	Hot-water packing for pump plungers, etc	0.20
†(*l*) *Special ice-machine packing:* Soft, rectangular, L. P. spiral packing of rubber and duck.	Stuffing boxes of expander and compressor rods	0.18
(*m*) *Asbestos wick or rope packing,* loosely twisted pure asbestos with about 15 per cent. cotton.	Use where high heat only (and no motion or movement) is to be resisted.	
(*n*) *Corset lacing or condenser-tube packing.*	Cotton cord for packing ends of condenser tubes, etc.	

It will be noted that (*b*), (*d*) and (*e*) are very expensive and their use is seldom absolutely necessary. The use of (*c*) may be desirable, since no waste material is left over due to cutting out gaskets.

All prices in the foregoing tables are per yard or square yard (not per pound), in order to show actual comparative values of cut gaskets or the quantity necessary to pack a stuffing box. Due to wide variations in weights of different brands, comparisons of "per pound" prices do not mean very much, though more convenient, since the weight of any amount of packing is easily and quickly obtained, and hence most packings are sold by the pound. The prices shown should be considered as comparative and in round numbers; they cover prices paid for very large amounts, and good packings at retail would cost considerably more.

(10) SMALL VARIETY OF PACKINGS AND SIZES DESIRABLE.

The list of navy contract packings comprises practically all kinds needed for the diversified machinery installations ashore and afloat. It might include a great many more kinds, but this would be very objectionable. In fact, there should be chosen from the contract list for each particular machinery plant a few good types, best suited to local needs, with sizes or dimensions limited in variety.

By sticking to a few good kinds, the mechanical force soon learns all about them, with consequent increased efficiency. Also this helps to eliminate "dead stock" and to provide at all times a good supply of new packings easily replenished upon short notice.

(11) SPECIAL PACKING AND JOINT MATERIALS.

Turbine casing joints are generally made up with some form of cement or wash (most frequently manganesite). Sheet packings in such a case are hard to set up evenly all around, if too thick increase the blading clearance undesirably, and the packing is likely to be squeezed out and protrude inside the casing.

Cement compounds, similar to "Smooth-on," are frequently used when making metal-to-metal joints of steel or iron.

Cup leather packings are used for compressor and expander pistons of ice machines and piston rods of oil cylinders of reversing engines.

For water plungers of pumps tucks or alternate rings of square flax and round tucks packings have been used with success.

Boiler-feed pumps may require a special hard tucks, cut in jointed rings like a piston ring, or solid white-metal plungers in order to properly handle very hot feed water.

Main steam-line slip joints and steam glands of pump rods should be packed with the regular high-grade spiral rod packing, which is most generally used, is always on hand or easily obtainable, and gives better results than special unusual types. When a steam-line slip joint is short and not deep enough to contain many rings of packing, it may be necessary to force in a few rings of metallic packing to support the joint and add a ring or two of the softer packing to prevent steam leakage.

Raw-hide packings are not considered desirable except in special cases, since they generally soften in water and do not wear well around a moving rod or shaft.

Internal-combustion engines may use the regular contract packings, but where there are a large number of small gaskets of the same size, as under the valve bonnets, it appears desirable to use made-up gaskets ready for use of metal and asbestos. The common type is a ring of sheet copper folded to give a U-section with asbestos inside.

Boiler blow valves of the outside-packed plunger type require hard, firmly compressed packing rings made mostly of asbestos.

Boiler gage glass grommets must be of a special composition, molded to the proper dimensions, which will resist steam and boiler water.

Special molded rubber gaskets are used for watertight hatches, doors and fixtures. They are made of a composition similar to the rubber sheet packings.

(12) TYPES OF BOILER GASKETS.

Gaskets of woven asbestos cloth, with brass or copper wire insertion, seamless and not jointed with tape, folded to a two, three or four-ply thickness, and with the layers *firmly cemented together by a high-grade binder* or "friction," are the only type which may be expected to give general satisfaction in all boilers. In other words, this type of gasket might be adopted as the standard handhole and manhole gasket for all boilers, provided sufficient properly made gaskets were obtainable.

To avoid repetition, the above will be hereafter referred to as the "standard" gasket.

Compressed asbestos-fiber gaskets (cut from a high-grade sheet packing) give good service in new installations or where joint surfaces are smooth. In fact, they may last longer than the "standard," if well made, and should be cheaper.

Handhole gaskets, which have to be made to fit narrow flanged joint surfaces, as for B. & W. boilers, will not prove satisfactory unless they are one of the two types above mentioned.

Asbestos cloth, jointed gaskets, the joint wrapped with tape, but otherwise similar to the "standard," are less expensive and are suitable for use where the joint has a broad surface; for example, for manholes of steam drums, mud drums, etc.

Metal and asbestos gaskets, if properly made, may prove very satisfactory except for handholes. They are built up of alternate layers of metal and asbestos firmly pressed together, are not flexible and, consequently, must be a proper fit. Copper is a metal frequently used, but is likely to be corroded by boiler water. The metal should be non-corrosive, not liable to become brittle in use and the gasket needs to be so constructed that the layers are firmly held together. The advantages of gaskets of this type are that they may be used over many times with careful handling, fewer spares need be carried, and they do not deteriorate in store.

Navy boiler gaskets comprise "standard" B. & W. square

and oval handhole gaskets, three sizes of handhole gaskets for Heine boilers, "standard" B. & W. manhole and mud-drum gaskets, metal and asbestos manhole and mud-drum gaskets for B. & W. boilers, and woven asbestos type gaskets of odd sizes. The latter are made up according to templates and have to be ordered from the contractor in advance of the need for them. Before being contracted for all brands of boiler gaskets must be thoroughly tested and found acceptable.

(13) SIMPLE TESTS FOR BOILER GASKETS.

A good gasket will stand boiling in fresh water for several hours without showing apparent change. Any metal in the gasket should not show signs of corrosion when partly immersed in distilled water or in brackish water for a few days. When a cement or binder is used, as in the woven asbestos-cloth type, it should be difficult to pull the gasket apart by hand, and there should not appear on the outside a coating similar to putty or white lead which may easily be scraped off. When baked at about 200 degrees F. for six hours, the gasket should not harden and lose its original properties.

(14) EFFECTS ON GASKETS OF BOILER COMPOUND, DISTILLED WATER AND HEAT.

Even a large quantity of Navy boiler compound in boiler water does not eat up good gaskets nor injure them, though there is a popular opinion to the contrary. Boiler compound might easily cause leaks, if there were rust or scale on either side of the gasket in the joint, since the compound attacks and dissolves iron rust or scale deposits such as frequently occur in boilers. However, hot distilled water does cause gradual deterioration, which is more marked in case the gasket has been handled roughly or set up too hard before the cement binder has had a chance to set under heat.

Dry heat will cause the rubber cement to soften and become sticky at first, then later to harden and become brittle. Naturally, "standard" gaskets kept in a hot place for a long period will be found to be unserviceable.

(15) USING GASKETS.

A thin gasket is best where joint surfaces are smooth and is less likely to blow out than a thick one.

Gasket seats should be well cleaned and pieces of old gaskets removed before placing the new gasket in position.

When installing a gasket, the object should be to apply, *when setting up on it, an even, well-distributed pressure* over as large a part of the surface of the gasket as possible. *Nuts should not be set up too tightly in the beginning,* but should be followed up as found necessary to prevent leaks when the steam pressure rises.

Cases have been experienced where handhole plates were set up so tightly with a long-handled wrench that the new gasket was mostly squeezed out of the joint and its structure torn in places. These small crevices or torn places form small openings, through which steam soon leaks, and the only remedy is to replace the gasket which has been ruined.

Gaskets with too wide a flange surface, or carelessly put in place, may project too far beyond the edge of the joint and jam the handhole or manhole cover at that point, so it cannot be set up or made to seat properly.

Gaskets of compressed fiber sheet are likely to tear when the joint is broken and to leave pieces of the old gasket sticking to the metal. These often form high spots which are easily overlooked, and likely to cause leaks and the condemnation of new gaskets which are not faulty.

APPENDIX.

INFORMATION CONCERNING TESTS OF PACKINGS AT THE ENGINEERING EXPERIMENT STATION.

ISSUED BY THE BUREAU OF STEAM ENGINEERING, NAVY DEPARTMENT. SUBJECT TO CHANGE OR MODIFICATION AS FOUND DESIRABLE. MARCH 31, 1917.

1. ***Sheet packings and gland packings, which are not bought under standard leaflet specifications*** and which are not known to be suitable for use in the naval service, ***must be tested at the Engineering Experiment***

Station, Annapolis, Md., before being considered acceptable by the Bureau of Steam Engineering.

2. Tests made at the Engineering Experiment Station must be authorized by the Bureau of Steam Engineering; and it is therefore necessary to make application in writing to that Bureau whenever it is desired to have any brand of packing tested.

3. *All tests* are authorized *with the strict understanding that they are for the information of the Government only; that they have to be made without expense to the Government;* and that *results obtained from tests will not be used in any way for advertisement.*

4. When a test is authorized, instructions are issued to the Experiment Station, and information is furnished the exhibitor in regard to the quality and quantity of material to be submitted, the estimated cost of making the test, and other preliminary details. Tests are numbered consecutively; and they are taken in hand in regular order, or as early as practicable after exhibits are received at the Experiment Station.

5. In addition to the requirement that *all expenses* of a test are to be *borne by the person or company making the application,* it is required that samples for test be furnished free of charge, that all express or other charges for shipment and delivery be prepaid, and that an amount sufficient to cover the *estimated cost of the test be deposited with the Superintendent, U. S. Naval Academy,* before the test is taken in hand. *Any unexpended balance* at the completion of test *will be returned* to the depositor.

6. Unless otherwise directed, *samples for test* are to be shipped for delivery to the "Supply Officer, U. S. Naval Academy, Annapolis, Md.," and each package should be *marked "For test at Experiment Station."*

7. All packings are subjected to practical tests, the steam pressures being the highest and other conditions the severest which the packings may be called upon to withstand in actual service. As the relative value of a packing depends chiefly upon its durability in actual service, each packing is continued under test until it is shown that it can no longer satisfactorily withstand test conditions or that it is capable of withstanding the test conditions for a reasonable length of time.

HIGH-PRESSURE PACKINGS.

8. *Sheet packings, for general use in high-pressure steam joints,* are tested in the form of gaskets in flange joints of the testing apparatus, which is made up of a number of short lengths of 6-inch pipe. Four gaskets of each brand of packing are put in different joints and are subjected to superheated steam at 300 pounds' pressure and a maximum temperature of 625 degrees F. for 8 hours each working day. During the remaining 16 hours of each day, and also on holidays, steam is shut off and the apparatus is allowed to cool to atmospheric temperature. The gaskets are subjected to this treatment for at least 200 working days, if they will survive treatment for that length of time.

9. For this test there will be required two square yards, 1/16-inch thick,

of each brand of sheet packing to be tested. Seven brands of packing can be tested collectively; and the cost of making the test will be $230.00 when only one brand is tested; or $40.00 for each brand when divided proportionately among seven brands.

10. *Sheet packings for high-pressure gas-engine joints* are tested in the form of gaskets in cylinder-head joints of internal-combustion engines. They are subjected to regular working conditions, and records are kept by which the durability and relative value of each brand of packing can be determined. The gaskets are subjected to this treatment for at least 100 working days.

11. For this test there will be required of each brand of packing to be tested:

1 square yard sheet packing 1/32-inch thick, and
1 square yard sheet packing 1/16-inch thick.

The cost of making this test will be $45.00 for each brand of packing submitted for test.

12. *Spiral gland packings for high steam pressures* are tested in the stuffing boxes of the rod-packing test apparatus, and are subjected to working conditions under steam pressures up to 300 pounds per gage. Each stuffing box is packed, in the same manner as in actual service, with the number of turns of packing necessary to fill the stuffing box, and the packing is compressed by the gland sufficiently to prevent leakage. The rod is made to travel up and down through the stuffing boxes at a speed of about 100 feet per minute, and provision is made to admit steam to the stuffing boxes and then allow it to escape into the exhaust pipe for each double stroke of the rod. Each brand of packing is subjected to this treatment 8 hours a day for at least 100 working days.

Special metallic or semi-metallic packings are generally also given this test.

13. Eight brands of packing can be tested collectively; but, as some of the stuffing boxes are made for 5/8-inch packing and the others for 9/16-inch packing, the samples submitted may be either of these two sizes. For this test there will be required about 20 feet of each brand of packing, the size being stated by the Experiment Station when the test is about to be taken in hand. The cost of making the test will be $110.00 when only one brand of packing is tested and $25.00 for each brand of packing when eight brands are tested collectively.

LOW-PRESSURE PACKING.

14. *Sheet packings, for general use in low-pressure steam joints,* are tested in the form of gaskets in flange joints of the test apparatus, which is made up of 6-inch pipe, in short lengths, similar to the apparatus used in testing sheet packings for high pressures. Four gaskets of each brand of packing are put in different joints, and are subjected to steam at 180 pounds' pressure for 8 hours each working day. During the remaining 16 hours of each day, and also on holidays, steam is shut off and the appa-

ratus is allowed to cool to atmospheric temperature. The gaskets are subjected to this treatment for at least 200 working days.

15. For the test there are required two square yards of 1/16-inch of each brand of sheet packing. Seven brands can be tested collectively, and the cost of making tests will be $140.00 when only one brand is tested; or $30.00 for each brand when divided proportionally among seven brands.

TUCK'S PACKING AND BRAIDED FLAX PACKING.

16. Tuck's packing and braided flax packing in alternate layers are tested in stuffing boxes of piston rods and plungers in water cylinders of feed pumps working against pressures up to 340 pounds. The stuffing boxes are packed with samples sent for test, and they are then subjected to regular working conditions for about 8 hours each working day, and, at times, continuously for periods of two to three weeks. The test is continued for at least 200 days.

17. For this test there will be required about 20 feet of Tuck's packing (round) and 20 feet of braided flax packing (square), each to be of the ⅝-inch size, unless otherwise stated. The cost of making the test will be $140.00.

GLAND PACKING FOR ICE MACHINES.

18. *Packings for use in stuffing boxes of compressor and expander cylinders of ice machines* are tested at the Experiment Station in the stuffing boxes of a ½-ton Allen dense-air ice machine by subjecting them to actual working conditions. This machine is kept running about 8 hours each day, and the test of each brand of packing is continued for at least 100 working days.

19. For the test two sizes of packing are needed: about 15 feet of 1/2-inch size and 15 feet of ⅜-inch size. The cost of making the test is $100.00.

MISCELLANEOUS.

20. Packings for miscellaneous purposes are subjected to such tests as will best determine the suitability of the packing for the purposes intended. In general, the conditions under which the samples are tested are similar to those to which they are to be subjected in actual service. These conditions can best be described when tests are being planned and it is known for what purpose the sample of packing is offered.

21. The quantity of packing required for test, the probable cost of each test, and information concerning other details will be furnished upon application to the Engineering Experiment Station.

22. It is desirable that the trade name of the packing furnished the Engineering Experiment Station for the original test be the same as that which it is intended to use in connection with bids, etc.; and that it be such as cannot be easily confused with the name of some other packing.

23. When a contract is awarded, the packings furnished on contract must be identical with samples sent to the Engineering Experiment Station for the original acceptance tests.

24. As correspondence and arrangements in connection with tests, the actual conduct of tests and unforeseen contingencies frequently consume many months, *prospective bidders should submit their packings* for test *eight or ten months before they intend to bid* for Government contracts.

INFORMATION CONCERNING TESTS OF BOILER GASKETS AT THE ENGINEERING EXPERIMENT STATION.

ISSUED BY THE BUREAU OF STEAM ENGINEERING, NAVY DEPARTMENT,
May 6, 1916.

SUBJECT TO CHANGE OR MODIFICATION AS FOUND DESIRABLE.

1. *All brands of boiler gaskets* not known to be suitable for use in the naval service, *must be subjected to test at the Engineering Experiment Station,* Annapolis, Md., and to further test in service in one or more naval vessels, before being considered acceptable by the Bureau of Steam Engineering.

(2, 3, 4, 5, same as "Tests of Packings," page 466.)

6. The tests to which samples of boiler gaskets are subjected at the Experiment Station are of a practical nature and such as will best determine the suitability of the gaskets for use in service. The conditions are made as severe as any which the gaskets may be expected to withstand in actual service. The cost of tests varies with the number of different brands of samples tested at one time, the labor involved, and the length of time required to make the tests; the expense being reduced where samples prove to be unacceptable in the early stages of the test.

7. Unless otherwise directed, *samples for test* are to be shipped to the "Supply Officer, U. S. Naval Academy, Annapolis, Md.," and each package should be marked "For Test at Engineering Experiment Station."

MANHOLE GASKETS.

8. All boiler manhole gaskets submitted for test at the Experiment Station are subjected to working conditions, with steam pressures up to 300 pounds, in manhole joints of a Babcock and Wilcox boiler, a Mosher boiler, or in special steam drums provided for the purpose. The boilers and steam drums are kept under working pressure at least 8 hours each working day, the pressure being brought up gradually in the morning and allowed to fall under banked fires at night. Samples of gaskets while undergoing test are subjected to these conditions daily until it is evident that they can no longer maintain tight joints, or until they show by endurance that they can satisfactorily withstand high steam pressures for a reasonable length of time. Manhole gaskets are subjected to this treatment for at least 100 days.

9. For the test there are required two oval manhole gaskets of each brand to be tested 15 inches by 11 inches by 1 inch flange by ⅛-inch thick,

suitable for steam pressures up to 300 pounds. The cost of making the test is $70.00 for one brand or style of gasket, and $40.00 for each brand when two brands are tested at the same time.

BOILER HANDHOLE GASKETS.

10. All handhole gaskets submitted for test at the Experiment Station are put in handhole joints of a Babcock and Wilcox boiler and subjected to regular working conditions with steam pressures up to 300 pounds. During the test the boiler is kept under working pressure at least 8 hours each working day, the pressure being brought up gradually in the morning and allowed to fall under banked fires at night. The samples of gaskets are subjected to these conditions daily until it is evident that they can no longer maintain tight joints, or until they show by endurance that they can satisfactorily withstand high steam pressures for a reasonable length of time. Each brand or style of handhole gaskets is subjected to this treatment for at least 100 days.

11. For this test there are required 30 handhole gaskets of each brand, square with round corners to suit Babcock and Wilcox marine boilers, 4-31/32 inches by 4-25/32 inches by 1/4-inch flange by 1/8-inch thick, adapted for steam pressures up to 300 pounds. The cost of making the test is $70.00 for one brand or style of handhole gasket; $40.00 for each brand when two or more brands are tested at one time.

(12, 13 and 14 same as paragraphs 22, 23 and 24, of "Tests of Packings," page 468.)

NAVY DEPARTMENT SPECIFICATIONS, 33P2B.

May 1, 1917.

ASBESTOS METALLIC CLOTH SHEET PACKING FOR HIGH-PRESSURE STEAM.

GENERAL SPECIFICATIONS.—1. General Specifications for the Inspection of Material, issued by the Navy Department, in effect at date of opening bids, shall form part of these specifications.

MATERIAL.—2. The cloth to be woven from best quality of long fiber asbestos yarn, spun around fine brass wire, and treated with a rubber or other suitable heat-resisting compound not exceeding 50 per cent. of the weight of the packing.

CONSTRUCTION.—3. For cloth 1/32 inch thick each strand of the warp and of the filler shall consist of one strand of asbestos yarn which is intertwisted with or spun around a brass or copper wire not heavier than 8 mils diameter. The weave must not run less than 22 strands of warp and not less than 12 strands of filler per square inch. For cloth 1/16 inch thick each strand of the warp and of the filler shall consist of two asbestos yarns, each of which is intertwisted with or spun around two brass or copper wires not heavier than 8 mils diameter. The weave must not run

less than 20 strands of warp and not less than 11 strands of filler per square inch.

FRICTION CEMENT.—4. One-ply packing to be 1/32 inch or 1/16 inch thick. Two-ply or more must consist of two or more layers of one-ply 1/16 inch sheet built up and thoroughly cemented together with the same compound as is used in frictioning one-ply cloth.

WEIGHT.—5. The weight of this sheet packing shall be not less per square yard than specified below in various thicknesses; a tolerance of 10 per cent. above these weights to be allowed: 1/32 inch thick, 5 pounds per square yard; 1/16 inch thick, 6 1/2 pounds per square yard; 1/8 inch thick, 12 pounds per square yard; 1/4 inch thick, 24 pounds per square yard.

DELIVERY.—6. This sheet packing shall be delivered in rolls 40 inches wide and weighing 100 pounds and 250 pounds unless otherwise ordered. To be purchased by the pound, the area to be calculated from the preceding paragraph (5), and shall be approximately square yardage so calculated.

BRANDING.—7. Each roll shall be marked with the maker's name, trade name or style number, neatly stenciled upon each linear yard.

TESTS.—8. (*a*) Shall be capable of being bent double in any direction without injury or showing signs of cracking.

(*b*) When secured in iron clamps and subjected for 24 hours to a steam heat of 425 degrees F. it must not harden or crack.

NAVY DEPARTMENT SPECIFICATIONS, 33P8B.

November 1, 1915.

PACKING, SHEET, RUBBER, CLOTH-INSERTION.

GENERAL INSTRUCTIONS.—1. "General Specifications for the Inspection of Material" and "General Specifications for Rubber Material," issued by the Navy Department, in effect at date of opening of bids, shall form part of these specifications.

CONSTRUCTION.—2. (*a*) *Packing in general.*—The packing shall be made of layers of rubber with cotton sheeting insertion, so arranged that both faces of the packing shall be a layer of rubber. There shall be a layer of rubber between each two layers of insertion. Both faces shall be free from imperfections and of uniform thickness. Packing shall be made in sheets 36 inches wide, and furnished in rolls containing not less than 100 pounds each, unless a smaller quantity is specified on the proposal.

For each 1/16 inch of thickness there shall be at least one ply of cotton insertion, which shall be frictioned with a rubber composition described below. If 1/32 inch packing is required it shall have but one ply of insertion.

(*b*) *Marking.*—Each roll shall be marked with the maker's name, date of manufacture, and number of square yards.

GRADES.—3. There shall be two grades of packing, known as Grade I and Grade II.

MATERIAL FOR GRADE I.—4. (*a*) *Rubber parts.*—The rubber layers shall be poperly vulcanized, and be made from and have all the characteristics of a compound containing not less than 35 per cent. of washed and dried fine Para rubber; not more than 2.5 per cent. of sulphur, with the remainder suitable, dry, inorganic mineral fillers. The mineral fillers may contain barytes, but shall be practically free from sulphur in other forms and from any substance likely to have a deleterious effect on the rubber compound. The sulphur in barytes will not be included in the allowable sulphur content. The specific gravity of the compound shall be not less than 1.7.

(*b*) *Friction.*—The friction shall consist of a properly vulcanized rubber compound, which will neither yield to acetone any organic constituent foreign to Hevea rubbers nor contain more sulphur than is necessary for vulcanization, so that the percentage of sulphur in the rubber layers shall not be raised beyond the permissible amount.

(*c*) *Cloth insertion.*—The cloth insertion shall be made of cotton sheeting, 30 to 40 threads per inch in both warp and filling, to weigh not less than 4¼ ounces per square yard. It shall be well and evenly woven from good cotton, and as free from unsightly defects, dirt, knots, lumps, and irregularities of twist as is consistent with the best manufacturing practice.

TESTS FOR GRADE I.—5. Grade I packings shall stand the following tests:

(*a*) *Friction test.*—The adhesion of the friction coat of rubber between the faces and the insertion shall be such that a weight of 9 pounds shall not cause a separation at a greater rate than 1 inch per minute. The width of the piece under test shall be 1 inch.

(*b*) *Tensile test.*—When the test piece is stretched 2 to 10 inches, the permanent elongation when measured shall not exceed 20 per cent. The tensile strength shall be at least 1,000 pounds per square inch in both the face and filler. Both shall have an elongation at the breaking point of at least 2 to 11 inches.

(*c*) *Test for 1/32-inch packing.*—If 1/32-inch packing is required, it shall not be subject to physical tests, but shall be accepted on chemical analysis alone.

(*d*) *Test pieces.*—The test pieces shall be cut by a die measuring ½ inch in width at the constricted part.

MATERIAL FOR GRADE II.—6. (*a*) *Rubber parts.*—The rubber composition layers shall be of material that will withstand the tests prescribed for this grade of packing.

(*b*) *Cloth insertion.*—The cloth insertion shall be made of cotton sheeting weighing not less than 5 ounces per yard. It shall be woven from good cotton, as free from unsightly defects, dirt, knots, lumps, and irregularities as is consistent with the best manufacturing practice.

(*c*) *Weight.*—The 1/16-inch complete packing shall weigh not less than

5 nor more than 6 pounds per square yard, and other thicknesses in the same proportion. Packing weighing over 6 pounds per square yard may be rejected, or accepted with payments based on area corresponding to 6 pounds per square yard.

TESTS FOR GRADE II.—7. Grade II packings shall withstand the following tests:

(*a*) The packing shall be pliable enough to stand bending double in any direction without cracking.

(*b*) A section of packing shall be placed on the flange of an autoclave so that the steel plate is clamped over the whole piece of packing and exposed to steam at 50 pounds per square inch pressure for 4 hours. After removal from the autoclave the packing shall stand being bent double in any direction without cracking or breaking when tested within 1 hour after cooling to room temperature and shall show no apparent injury.

(*c*) A section of packing 4 inches square shall be placed in the water of the autoclave and the autoclave heated by steam at a pressure of 50 pounds per square inch for 4 hours. After removal from the autoclave the packing shall stand being bent double in any direction without cracking or breaking when tested within an hour after cooling to room temperature and shall show no apparent injury.

(*d*) Packing ⅛ inch in thickness and over shall not be required to be bent double, but shall instead stand bending round a mandrel ½ inch in diameter.

PURPOSE.—8. Grade I packing is suitable for use as high-grade gasket material in flanges of ventilation systems or flanges in contact with cold water which are not broken except at long intervals.

Grade II packing is suitable for use as gasket material in flanges in contact with cold water which are broken at frequent intervals.

NOTE FOR GENERAL STOREKEEPERS.—9. In ordering packing the requisition shall distinctly state whether Grade I or Grade II packing is desired.

NAVY DEPARTMENT SPECIFICATIONS, 33R1b.

December 1, 1915.

RUBBER: MOLDED, SHEET, AND STRIP.

GENERAL SPECIFICATIONS.—1. "General Specifications for Inspection of Rubber Material" and "General Specifications for Inspection of Material," issued by the Navy Department, in effect at date of opening bids, shall form part of these specifications.

PURPOSE.—2. To be used for gaskets and as water-tight packing for doors, hatches, gunports, etc.

GENERAL.—3. (*a*) *Sheet rubber* shall be of uniform thickness and have a smooth finish. It shall be furnished in rolls 36 inches wide, containing not less than 100 pounds each, unless a smaller quantity is specified in the proposal.

(*b*) *Strip rubber* shall be of uniform thickness, shall have a smooth finish, and shall be of the dimensions specified.

(*c*) *Molded rubber* shall be of the shape and dimensions specified. Molds shall be furnished by the contractor, unless otherwise stated in the proposal.

TOLERANCES.—4. A variation of 10 per cent. above the thickness specified will be permitted for sheet and strip rubber; provided, that such increase shall not exceed 1/16 inch.

MATERIAL.—5. The gaskets shall be properly vulcanized and be made from and have all the characteristics of a compound containing not less than 45 per cent. of washed and dried fine Para rubber, not more than 3.5 per cent. of sulphur, with the remainder suitable, dry, inorganic mineral fillers. The mineral fillers may contain barytes, but shall be practically free from sulphur in other forms and from any substance likely to have a deleterious effect on the rubber compound. The sulphur in barytes will not be included in the allowable sulphur content.

TESTS.—6. (*d*) With each delivery of molded rubber, the contractor shall furnish four sample pieces 8 inches long, 1½ inches wide and ⅛ inch thick. They shall be guaranteed by the manufacturer to be identical in composition and cure with the rubber offered for delivery, and on analysis shall show the same chemical composition as the delivery.

Test pieces shall be cut from these samples. In the case of sheet and strip rubber, the test pieces shall be cut from the delivery itself. All test pieces shall be cut by a die measuring ¼ inch in width at the constricted part.

(*e*) When the test piece is stretched 2 to 11 inches, the permanent elongation, when measured, shall not exceed 20 per cent. The tensile strength shall be at least 2,200 pounds per square inch, and the elongation at the breaking point shall be at least 2 to 12 inches.

MARKING.—7. Each roll or package shall be marked with the maker's name, the date of manufacture, and the net weight in pounds.

NAVY DEPARTMENT SPECIFICATIONS, 33P1.

August 15, 1908.

FLAX PACKING.

Flax packing to be square and of the sizes required.

To be made from the best selected long-line Russian flax, well and evenly impregnated with pure tallow.

To be free from warp or core; from flax tow, jute, or other foreign fibers or substances, and from oils and greases other than pure tallow, which latter must not be in excess of 50 per cent., by weight, of finished packing.

The length of staple must be such that when a section of packing 20 inches long is carefully unraveled and combed out with a coarse comb it must give a brush of at least 12 inches.

NAVY DEPARTMENT SPECIFICATIONS, 33P7a.

July 25, 1913.

TUCK'S PACKING.

GENERAL INSTRUCTIONS.—1. General Specifications for Rubber Material, 33R3, issued by the Navy Department, shall form part of these specifications.

CONSTRUCTION.—2. (*a*) *Packing in general.*—The packing shall consist of frictioned cotton canvas layers, either round or square. The round packing shall be made with or without a rubber core, as specified on the original proposal.

COTTON CANVAS LAYERS.—(*b*) The cotton canvas layers, which shall be built up on the bias and shall be well frictioned with the composition described below under "MATERIAL," shall conform in dimensions, plies, and textures to the following table:

Sizes	¼ to ⅜ inch.	½ to ⅞ inch.	1 inch and over.
Texture of plies	Fine	Medium	Coarse.
Number of plies per inch, including friction	38 to 42	26 to 30	15 to 17.

RUBBER CORE.—(*c*) The rubber core shall have a diameter equal to 1/3 of the diameter of the packing.

MATERIAL.—3. (*a*) *Cotton canvas layers.*—The cotton canvas layers shall be well and evenly woven from good cotton, as free from unsightly defects, dirt, knots, lumps, and irregularities of twist as is consistent with the best manufacturing practice.

RUBBER CORE.—(*b*) The rubber core shall be properly vulcanized and be made from and have all the characteristics of a compound containing not less than 35 per cent. of washed and dried fine Para rubber, not more than 3 per cent. of sulphur, with the remainder suitable, dry, inorganic mineral fillers. The mineral fillers may contain barytes, but shall be practically free from sulphur in other forms and from any substance likely to have deleterious effect on the rubber compound. The sulphur in barytes will not be included in the allowable sulphur content.

FRICTION.—(*c*) The friction shall consist of a properly vulcanized rubber compound which will neither yield to acetone any organic constituent foreign to Hevea rubbers nor contain more sulphur than is necessary for vulcanization, so that the percentage of sulphur in the rubber layers shall not be raised beyond the permissible amount.

TESTS, HEAT.—4. *Tests on the packing as a whole.*—(*a*) The completed article shall stand boiling water with steam at 80 pounds' pressure for four hours without loosening the friction or materially hardening the rubber.

FLEXIBILITY.—(*b*) The packing shall be sufficiently flexible to admit of bending easily around rods or pistons of various sizes.

The ¼ inch to ¾ inch shall bend around its diameter; the ⅞ inch to 1⅛ inches around twice its diameter, and the larger sizes around three times their diameter.

TESTS OF THE RUBBER COMPOUND.—(*c*) A section 6 inches long shall be cut from any length at any place to allow of four or more tests.

FRICTION.—(*d*) The adhesion of the friction coat of rubber between the plies shall be such that a weight of 8 pounds shall not cause the plies to separate at a greater rate than 1 inch per minute. The width of the piece under test shall be 1 inch, if size of the packing permits. In case of square packing, if the width of the test piece is less than 1 inch, the weight required shall be in direct proportion to its width.

RUBBER CORE. TEST PIECE.—(*e*) The round rubber core of 3/16 inch in diameter or under shall be used as the test piece. If the core is larger than 3/16 inch in diameter, it shall be cut as described in General Specifications 33R3.

TENSILE STRENGTH.—(*f*) The tensile strength shall be at least 900 pounds per square inch, and it shall have an elongation at the breaking point of at least 2 to 8 inches.

PURPOSE.—5. For use as stuffing-box packing, on piston in pumps and engines, etc.

NAVY DEPARTMENT SPECIFICATIONS, 33P5d.

January 2, 1917.

ASBESTOS WICK OR ROPE PACKING.

GENERAL INSTRUCTIONS.—1. General Specifications for Inspection of Material, issued by the Navy Department, in effect at date of opening of bids, shall form part of these specifications.

MATERIAL.—2. To be made from at least 90 per cent. long fiber asbestos spun into yarns and laid up in the general form of loosely twisted cord, or common lamp wick, which can be separated and rewound into grommets or braided into small gaskets.

SIZES.—3. The packing will be in sizes from ⅛ to 1¼ inches.

PACKING.—4. Packing will be delivered in usual commercial forms or containers, and will be protected from injury in handling or shipping. Packages should be marked with the manufacturer's name and brand or style number.

TESTS.—5. (*a*) Under the flame of a lamp applied with a blowpipe, it should not burn nor emit flame.

(*b*) Dipped in undiluted muriatic acid, it should not burn or disintegrate.

NAVY DEPARTMENT SPECIFICATIONS, 33P9a.

May 1, 1914.

PACKING FOR CONDENSER TUBES.

GENERAL INSTRUCTIONS.—1. General Instructions or Specifications issued by the Bureau concerned shall form part of these specifications.

MATERIAL.—2. Condenser tube packing will be what is commercially known as cotton corset lacing.

MANUFACTURE.—3. To be made up of 20 strands of cotton woven around a 16-strand cotton core to an approximate diameter of ⅛ inch. The finished material to run 200 yards to the pound.

SIZE AND TEXTURE.—4. To be of uniform texture and diameter. In all respects suitable for packing Navy standard condenser tubes.

SAMPLES TO BE SUBMITTED.—5. Bidders must submit sample of material that they propose to furnish.

DELIVERY IN 1-POUND BALLS.—6. This packing is to be delivered in 1-pound balls of continuous length.

NAVY DEPARTMENT SPECIFICATIONS, 33L1a.

December 1, 1913.

PACKING LEATHERS FOR COMPRESSOR AND EXPANDER CYLINDERS OF DENSE-AIR ICE MACHINES.

GENERAL INSTRUCTIONS.—1. General Instructions or Specifications issued by the Bureau concerned shall form a part of these specifications.

MATERIAL.—2. To be made of best grade leather specially treated to withstand heat.

PACKING.—3. To be saturated with castor oil and packed in airtight tin cans, one dozen leathers in each can. Cans to be well filled with cold-pressed castor oil and to be plainly marked to show contents, size, and name of contractor.

TESTS.—4. Sample leathers from each lot will be tested by subjecting them to a temperature of 350 degrees F., and by soaking in boiling water, after which they shall remain pliable and suitable for use.

SIZES.—5. Leathers to be made accurately to the sizes in the following table:

Ice machine.	Cylinder.	Outside diameter. Inches.	Inside diameter. Inches.	Thickness. Inches.
½ ton vertical	Compressor	5⅜	1¾	3/16
½ ton vertical	Expander	4⅝	1¾	3/16
1 ton vertical	Compressor	7¼	2½	3/16
1 ton vertical	Expander	6⅛	2½	3/16
1 ton horizontal	Compressor	6⅜	1⅞	¼
1 ton horizontal	Expander	5⅜	1⅞	¼
2 tons horizontal	Compressor	7¾	3	¼
2 tons horizontal	Expander	6½	3	¼
3 tons horizontal	Compressor	9¾	3¼	¼
3 tons horizontal	Expander	8⅛	3¼	¼

NAVY DEPARTMENT SPECIFICATIONS, 33G1b.

February 1, 1915.

GAGE-GLASS GROMMETS.

GENERAL INSTRUCTIONS.—1. General specifications for the inspection of material issued by the Navy Department in effect at date of opening of bid shall form part of these specifications.

CONSTRUCTION.—2. The grommets shall be either cut from a vulcanized

tube of solid rubber or molded from an unvulcanized compound, both of which shall be of the composition specified below under "Material." They shall be smooth and uniform in finish, rectangular in cross section, 5/16 inch in length, and of the external and internal diameters specified in the original proposal.

MATERIAL.—3. The grommets shall be properly vulcanized and be made from and have all the characteristics of a compound containing not less than 35 per cent. of washed and dried fine Para rubber, not more than 2.5 per cent. of sulphur, with the remainder suitable, dry, inorganic, mineral fillers. The mineral fillers may contain barytes, but shall be practically free from sulphur in other forms and from any substance likely to have a deleterious effect on the rubber compound. The sulphur in barytes will not be included in the allowable suphur content.

TEST PIECES.—4. With each delivery the contractor shall furnish 4 sample pieces 8 inches long, 1½ inches wide, and 1/16 inch thick to provide for physical tests. They shall show on analysis the same composition as the grommets offered for delivery. From these samples test pieces shall be cut by a die measuring ½ inch in width at the constricted part.

TENSILE STRENGTH, ETC.—5. When the test piece is stretched 2 to 10 inches the permanent elongation when measured shall not exceed 25 per cent. The tensile strength shall be at least 1,500 pounds per square inch and the elongation at the breaking point at least 2 to 11 inches.

HEAT TEST.—6. The grommets must stand exposure to boiling water for four hours without apparent injury, and after exposure to steam at 250 pounds' pressure for four hours in an autoclave or in superheated steam must show a considerable degree of flexibility.

PURPOSE.—7. For use in packing water-gage glasses on boilers.

NAVY DEPARTMENT SPECIFICATIONS, 33P11.

September 1, 1916.

PACKING RINGS FOR BOILER BLOW-OFF VALVES.

GENERAL SPECIFICATIONS.—1. General Specifications for Inspection of Material, issued by the Navy Department, in effect at date of opening of bids, shall form part of these specifications.

PURPOSE.—2. The packing rings are to be used in boiler blow-off valves.

MATERIAL.—3. The rings to be made of 90 per cent asbestos with rubber as a binder. This material is to be treated with a compound and compressed to such a density as to make the rings sufficiently resilient to form a tight joint. The material is to be such as to make the rings sufficiently hard and tough to withstand the action of opening and closing the valve without tearing or roughening the exposed surface of the ring. The substance used as a binder must be such that it will not disintegrate at temperatures up to 500 degrees F.

CONSTRUCTION.—4. Each ring to be of the split type and the two sections of the ring are to closely fit each other when the concave side of one section and the convex side of the other section are placed together.

TESTS.—5. The material shall not burn or disintegrate when dipped in undiluted muriatic acid, and when secured in an iron clamp and subjected to the heat of steam at 300 pounds' pressure shall show no evidence of hardening, cracking, or sticking to the metal clamps.

MATERIAL.—3. The rings to be made of 90 per cent. asbestos with rubber trade name and name of manufacturer.

NOTE FOR SUPPLY OFFICERS.—The type of valve for which intended and all dimensions of the rings are to be carefully specified in the order, and rings will be in accordance with these dimensions.

AERONAUTICAL PROPELLERS.

By Spencer Heath.*

The aeronautical screw propeller or air screw, strange as it may seem, was, in point of theory and conception at least, the predecessor and forerunner of the marine screw propeller. In the year 1768 there was published in France a treatise on the "Theorie de la Vis d'Archemedes." This very early publication, written by one A. J. Paucton, describes a machine equipped with two screws called by the author "Pterophores," description of which occupies a considerable portion of the book. It is noteworthy that to France we owe the first publication of screw propeller theory as well as the first public flight of an aeroplane propelled by that means.

In their basic principle all screws are the same for whatever purpose used. They differ only in their proportions, form and material. These differences are due to the different nature of materials through which the screws act. A machine screw acting through a previously prepared nut is a special case of a member progressing along an inclined plane, the movement of the member being along a helical curve instead of a straight line. A wood screw and a corkscrew, however, make their own "nut" through the cork or wood in which they act. These screws, in common with the machine screw, have an axial advance in one revolution substantially equal to the distance between turns of the "thread," known as the pitch of the screw. The air and marine screw are like the cork and wood screw in that they form their own "nut" through the air or water, but they differ markedly in that the fluid medium through which they pass has no great stability

* President, American Propeller & Manufacturing Co.

and yields to the screw in such manner that commonly the screw does not advance its full pitch in one revolution. This yielding of the fluid is entirely analogous to the yielding of the water to the bending of an oar or paddle. The resistance with which the fluid yields in the sternward direction creates a fulcrum for the oar and measures the propulsive impulse of oar or screw. The magnitude of this impulse depends upon the density of the fluid medium and the sternward velocity with which it is caused to move.

The differences between air and marine screws may be said to be the reflex of the differences between air and water. Air is thinner, lighter, larger, swifter than water. Air screws are longer and thinner of blade, lighter in weight of material, larger in proportion to duty and swifter in velocity of rotation and of flight.

Almost all air screws are made of wood. In general appearance, except for width of blade, they resemble marine screws. The number of blades may be two, three or four. The widest part of the blade is usually at about six-tenths of its radius. The maximum width averages about one-tenth to one-fifteenth the diameter of the screw. The thickness of blade near the hub is very great, but diminishes rapidly to about half blade length and then gradually to the end. The side of the blade facing rearwardly is slightly concave in its wider and mid-radius portions and flat or slightly convex at its narrowed end and near the hub. The forward facing side of the blade is convex in all parts and the greatest thickness of any section is about one-third the distance from the entering to the trailing edge. Nearly all air screws have the general characteristics already mentioned. In blade outline, however, there is wide diversity. Some designers prefer to approximate a slender ellipse; others prefer the slender ellipse with squared ends on the blades; others approximate a semi-ellipse, the axis of the ellipse proceeding radially from the axis of rotation and forming the trailing edge of the blade. Among these various forms no special preference is known.

Some of them are laid out with a view to placing the centers of gravity of all the blade sections in one radial line; some with the aim of having the center of pressure on each blade section lie in the same straight line.

In all the above forms of blade a common property obtains: The deflection of the blade under load is accompanied by more or less increase of angle in its most effective parts, thus augmenting the pitch. This gives the blade a sort of unstable pitch which may introduce heavy strains and resistances to the turning of the propeller at the moment when highest turning speed is required. In order that the pitch may remain unaffected by bending or deflection of the blades under load or increase of load it is necessary to dispose the centers of pressure of the sections farthest from the hub on a line curving somewhat rearwardly in the direction of the trailing edge. This practice is followed to a large extent by the company with which the writer is connected, who also specialize very much in variable-pitch propellers. These are designed by so far extending the rearward curvature of the blades that the application of working blade pressure will institute a torsion action on the entire blade, causing its pitch to increase or diminish in response to variation in pressure. In this torsion design for variable pitch the portion of wood employed in the curved trailing edge of the blade is steamed and bent so that the grain of wood parallels the curved edge of blade. In this process a slight compression is given to the fibers which greatly increases the endurance of the thin-edge portions of the blade.

Nearly every kind of wood has been used in propellers. Walnut and mahogany have long been favorite in Europe. The writer's experience rules out all wood that was not quarter sawed, and points to American quartered white oak as the surpassing material from every standpoint, the particulars of which need not be detailed here.

It is almost needless to say that the wood for air screws is selected and treated with utmost care. The boards are sawn

to one inch, rough-dressed to seven-eighths and finish-dressed to thirteen-sixteenths or three-quarters.

The screws are built up by five to ten laminations according to size. The laminations are laid out on the boards and sawed to outline, care being taken to avoid all defects in wood and to have the grain and density of wood as nearly similar as may be at opposite ends of the same piece. In the better and preferred practice, however, the laminations for each single blade are laid out separately and carefully weighed, matched and balanced against each other. They are then selected in pairs (or in trios for three-bladed screws) and their hub ends securely glued together in highly efficient joints of very large glue-contact area. Only by this method is it possible to make the blades of the same propeller uniform in respect to weight, grain, texture and yielding of the wood under stress.

When the separate laminations have been prepared and surfaced to required thickness they are slightly roughened by tooth planing, warmed over steam coils and assembled together with the best of hide-stock glue and firmly clamped. The entire gluing process is carried on in a room kept at 100 degrees F. After eighteen to twenty-four hours the clamps are removed and center hole in hub bored roughly to size. The propeller is now hung for about ten days to dry. It is then put through a machine which at one operation faces both sides of the hub and bores out the center hole to finish size. After being faced and bored the propeller is "outlined" in a machine that profiles the hub and edges of the blades all to exact size and shape by means of a rotary cutter following a form which has the precise outline of the blades.

From the outlining machine the propeller progresses to one of the duplicators. In this machine the work is clamped in definite relation to a rigid, fixed form having the same shape as the blades. On the carriage of the machine there is a roller which traverses the surface of the form and guides a high-speed cutter in a manner to remove nearly all surplus

wood from the rough propeller. The carriage is self-feeding and self-reversing, and the bed and other parts of the machine, including the form, are made duplex in order to secure continuous operation of the cutter, the work being removed and renewed at each end of the machine in turn while the carriage is operating uninterruptedly at the other end.

After the duplicating process the propeller again dries for a few days, after which it is carefully surfaced and balanced by hand and then forwarded to the sanding machine. After sanding there is careful inspection in the white and careful examination of balance, pitch and tracking of blades, hub dimensions, etc. The inspection itself is an elaborate process requiring special appliances, etc., of various kinds.

From inspection the propeller passes to the finishing department. Here it is first treated with silex filler, then with primer, and lastly with various coats of high-test waterproof spar varnish. This is the usual finish. For certain United States Army work five applications of hot linseed oil and a final rubbing with prepared wax are required. During the entire varnishing process the propeller is kept carefully balanced on a steel mandrel resting on sensitive parallel ways. Without this very few propellers could pass final inspection.

On final inspection the utmost attention is given to every detail. Balance must be absolutely perfect in all positions; the blades must track within .03 inch; the pitch of the blades, checked at three points, must not vary from the standard by more than two per cent. nor from each other more than one per cent. These limits are only to allow for possible changes in the wood during the finishing process after inspection in the white.

A few words should be said as to the number of blades: For training work and all ordinary work, provided a sufficient diameter can be swung, two blades are usually preferred. For expert flying and for high-powered machines in which there is a restricted diameter of propeller in proportion to power applied, three and four blades are required. As to the

relative merits of three and four blades there is no conclusive data. It is known, however, that in numerous instances the three-bladed screw, even though having less diameter, shows marked superiority over the two-bladed *in every particular*. The three-bladed propeller is also noted for its peculiar jointing and fitting of the ends of the laminations together where they form the hub. This hub is trebly laminated over its entire area with the material so disposed as to direction of grain, etc., that it makes without doubt the strongest hub that can be built in any propeller regardless of the number of blades. In repeated cases of wreck and accident all the blades of these propellers have been wholly demolished, leaving the hubs always intact.

After final inspection the propellers are usually packed in standardized pine or white cypress boxes with screwed-on covers and heavy battens and iron-bound ends. A center bolt clamps the propeller between battens in the top and bottom of the box, and felt-lined pillow blocks, formed to the shape of the screw, secure it firmly in place.

Probably no essential instrument in aeronautics has more nearly reached its ultimate development than has the screw. However, the elastic nature of the air and its wide divergence from water in its other qualities opened up possibilities of development that could not enter into the marine propeller problem. This article, devoted mainly to constructional features, cannot go into the various departures from marine propeller practice that have been indicated by theory and experiment and confirmed in practice. These relate to the particular forms of blade sections preferred for the air, the ratio of blade and disc area, the relation between pitch and width in different parts of the same blade, and many other questions of great interest.

Occasionally the suggestion is made that the screw is not always to remain the chief propelling instrument for air craft of every sort, but that it will be finally displaced by some superior device. Many substitutes have been patented or pro-

posed, varying in form from tiny jets of steam, gas or air to huge devices like amusement wheels with suspended cars, but none of them, not even the ornithopter or wing-flapping device, has become a recorded success. The very high efficiencies reported for air screws under certain conditions (somewhat more than eighty per cent.), even if only moderately improved by future developing or refinement, present a none too encouraging outlook for the more complex and pretentious affairs that are frequently proposed.

MEASURING SHAFT HORSEPOWER OF DESTROYERS.

By Lt.-Comdr. F. W. Sterling, U. S. Navy, Member.

The shaft horsepower transmitted by a turbine-driven shaft is directly proportional to the torque in the shaft. Given a means of measuring this torque, the shaft horsepower can be determined. Since the torsion in the shaft varies as the torque applied to the shaft, the shaft horsepower may be determined if the torsion in the shaft is known. As its name implies, it is the function of the torsion meter to measure torsion. How this is done in a turbine shaft will be described later.

For destroyers a torsion meter must satisfy the following requirements:

1. Accuracy.—This is a fundamental requirement in all measuring instruments. Having once been calibrated, the torsion meter must measure accurately the torsion in the shaft under all conditions of load or speed within the range of the power unit during the life of the shaft.

2. Lightness.—Machinery weights in destroyers must be reduced to the lowest safe limit. This applies to all permanent fittings. Therefore any permanently installed torsion meter must combine lightness with accuracy. It is specially desirable that those parts of the instrument subject to centrifugal action be made as light as possible.

3. Balance.—Those parts of the instrument subject to centrifugal action must be in dynamic balance.

4. Simplicity of construction.—The instrument must be simple in construction and easy of operation, so that it may be used and maintained in good order by the average engineer force of a vessel.

5. There should be no gears or electrical attachments. The instrument must be positive in its performance; gears that may wear, or electrical equipment that may become inaccurate, introduce errors in the instrument that destroy its dependability.

6. The shaft torsion should be measured over the maximum length of shaft possible. The object of this is to reduce the action of the multiplying device as much as possible. By measuring the torsion in an entire length of shaft, only one multiplying device is necessary and its action is a minimum.

7. Space occupied should be a minimum. This is accomplished by housing most of the torsion meter within the shaft. Meters that surround the shaft have not proved satisfactory in the Service.

8. Accessibility.—All parts that require adjustment or that might get out of order must be accessible. They should be outside the shaft.

9. Those parts subject to wear should be reduced to the smallest possible number, and adjustments for wear should be easily and quickly made.

10. The personal equation must be eliminated; readings should be mechanically recorded, thus furnishing permanent records for future reference.

The above exacting requirements have been satisfied by one torsion meter on the market and it now enjoys the distinction of being standard equipment on U. S. Destroyers. On page 762 of these Proceedings, Volume 27, Number 4, Captain C. W. Dyson, U. S. Navy, says, in his article on The Mystery of the Screw Propeller, "In all the above cases [referring to a large number of trials on various types of naval vessels] where shaft horsepower was measured by means of torsion meters, the Gary-Cummings torsion meter was used and the results obtained speak volumes in favor of the accuracy of the instrument."

The Gary-Cummings torsion meter combines lightness with accuracy. Its construction, care, and operation, are simple and the personal equation is eliminated. No gears

or electrical equipment are present, and those parts subject to centrifugal action are very light and perfectly balanced. It measures the torsion over the entire length of one section of shaft, is mostly housed within the shaft, all parts requiring attention are easily accessible, and parts subject to wear are few and easily adjustable.

The method of measuring the torsion in destroyer shafts by this instrument and of determining the shaft horsepower therefrom will be described, together with the construction, operation, calibration and care of the instrument.

PRINCIPLE.

The relation between the shaft horsepower being transmitted in a shaft and the torque in the shaft is given in the formula

$$\text{S.H.P.} = \frac{2 \times \pi \times \text{M} \times \text{R.P.M.}}{33{,}000},$$

where

M = Torque or moment in foot pounds;
R.P.M. = Revolutions per minute.

The latter factor may be obtained by a tachometer or other independent means, the former by the torsion meter.

What the torsion meter actually does is to measure the torsion or twist that is present between the two coupling faces of one section of shaft when transmitting power. To better understand the operation, suppose the shaft could suddenly be brought to a state of rest, retaining the twist present when the shaft is transmitting power. It would be a simple matter to measure the twist or torsion, and from this and a calibration of the shaft determine the torque necessary to cause this torsion. The instrument is simply a device to measure this torsion while the machinery is in operation.

DESCRIPTION OF TORSION METER.

Meter Tube.—The torsion meter, as shown in the phantom view, Fig. 1, and the line sketch, Fig. 2, consists of a rigid steel tube housed in a complete section of shaft, secured

rigidly at one end but free to move at the other. At the free end is mounted the attachment for measuring the torsion between the two shaft flanges. The fixed end of the tube, Fig. 3, is rigidly secured, by riveting, to a collar on a spider which, in turn, is an absolutely tight fit inside the shaft. A radial arm is secured to the face of the spider by machine screws prick-punched to prevent backing out, and this radial arm fits tightly into a slot cut in the shaft flange and is secured thereto by bolts. This construction insures a positively fixed end.

The opposite end of the tube, Fig. 4, is free to move relative to the shaft. A collar is riveted on this end of the tube. This collar, in turn, rests in a ball bearing within a spider which is rigidly secured to the shaft and which forms a bearing rest for and centers the tube in the shaft. This spider is made to fit tightly in the shaft and is secured thereto by taper plugs in each spider foot which expand the four spider feet hard against the shaft. Two webs connecting adjacent feet of the spider add to its rigidity. The tube extends through the ball race within the spider and has secured across its free end a radial arm that extends through a slot in the shaft, the slot being of sufficient size to allow the radial arm to move freely in a plane perpendicular to the shaft axis. Part of the recording apparatus is secured to this radial arm, Fig. 4.

By assuming, when the shaft is free of torsion, a reference mark made on the shaft at the free end of the tube opposite the radial arm, it is obvious that, with one end of the tube fixed and the other end free to move relative to the shaft, any torsion in the shaft will be indicated by the angular displacement of the free radial arm from the reference mark. At the free end the meter tube is simply supported and centered. Its position is entirely controlled from its fixed end and its free end can rotate or move along the shaft as it expands or contracts. The relative position of the two flanges is what the meter measures and transmits to the recording device.

Recording Device—An umbrella-shaped device, equipped with two recording points, is so fastened to the outside of the shaft flange at the free end of the meter tube that its axis

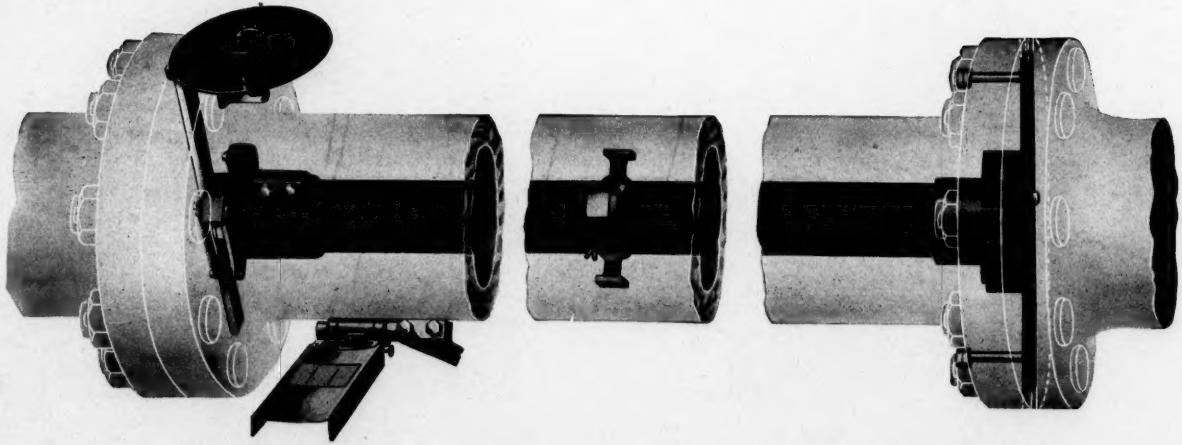


FIG. 1.—PHANTOM VIEW OF SHAFT SECTION WITH GARY-CUMMINGS TORSION METER INSTALLED.

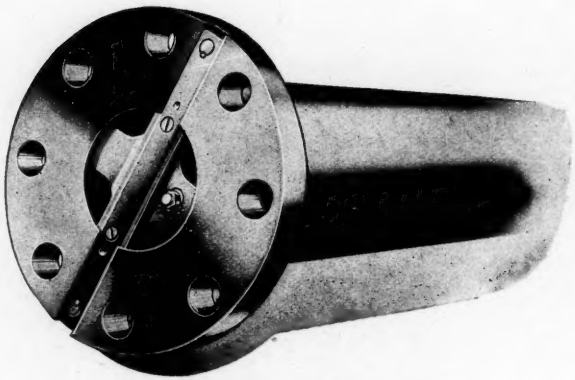


FIG. 3.—FIXED END OF TORSION METER.

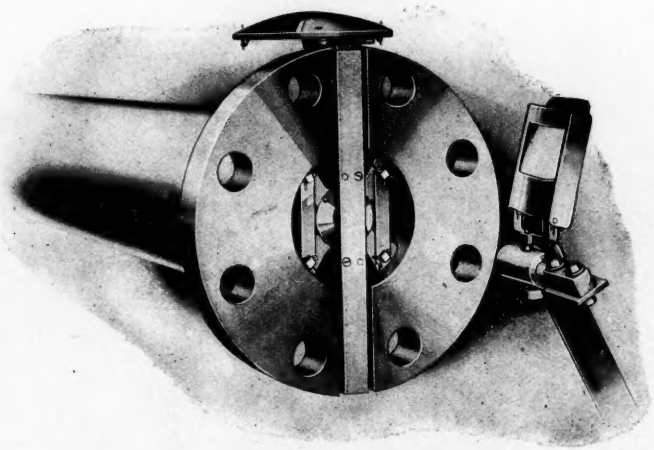


FIG. 4.—FREE OR RECORDING END OF TORSION METER.

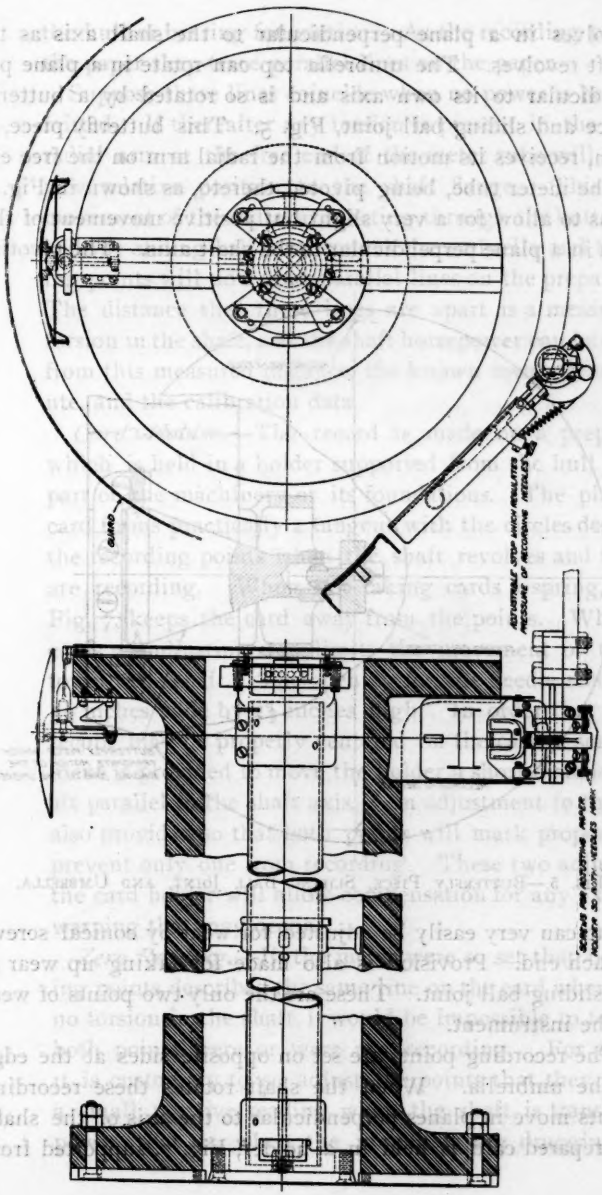


FIG. 2.—TORSION METER AND CARD-HOLDER.

revolves in a plane perpendicular to the shaft axis as the shaft revolves. The umbrella top can rotate in a plane perpendicular to its own axis and is so rotated by a butterfly piece and sliding ball joint, Fig. 5. This butterfly piece, in turn, receives its motion from the radial arm on the free end of the meter tube, being pivoted thereto, as shown in Fig. 6, so as to allow for a very slight but positive movement of this arm in a plane perpendicular to the shaft axis. The pivoting

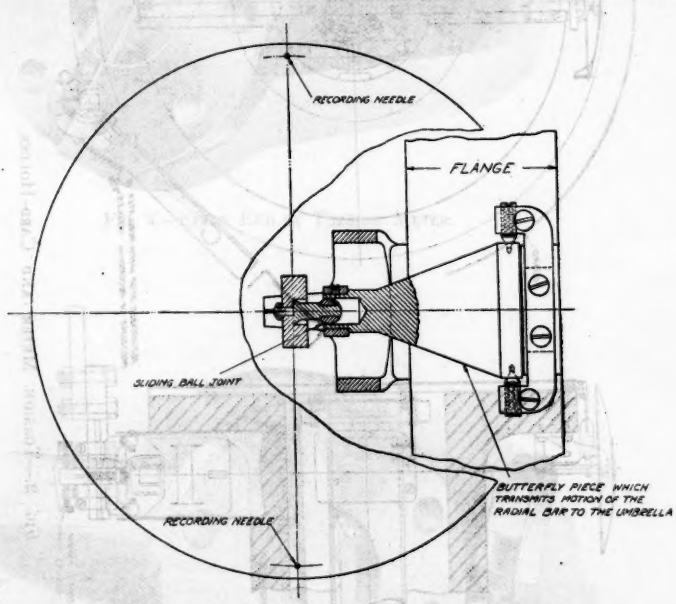

RECORDING NEEDLE
FLANGE
SLIDING BALL JOINT
BUTTERFLY PIECE WHICH TRANSMITS MOTION OF THE RADIAL BAR TO THE UMBRELLA
RECORDING NEEDLE

FIG. 5.—BUTTERFLY PIECE, SLIDING BALL JOINT, AND UMBRELLA.

point can very easily be adjusted for wear by conical screws at each end. Provision is also made for taking up wear at the sliding ball joint. These are the only two points of wear in the instrument.

The recording points are set on opposite sides at the edge of the umbrella. When the shaft rotates these recording points move in planes perpendicular to the axis of the shaft. A prepared card is held in a holder, Fig. 7, supported from

the hull or bearing foundation. As the recording points pass the paper they trace parallel lines on the paper.

Suppose these lines coincide when no power is being transmitted. If thereafter any torsion is present in the shaft, the radial arm at the free end of the meter tube will assume a new relative position to the shaft flange. This apparent movement of the radial arm acting through the butterfly piece and the sliding ball joint rotates the umbrella and the recording points will now draw parallel lines on the prepared paper. The distance that these lines are apart is a measure of the torsion in the shaft, and the shaft horsepower can be calculated from this measured distance, the known revolutions per minute, and the calibration data.

Card Holder.—The record is made on a prepared card which is held in a holder supported from the hull or a fixed part of the machinery or its foundations. The plane of the card forms practically a tangent with the circles described by the recording points when the shaft revolves and the points are recording. When not taking cards a spring, shown in Fig. 7, keeps the card away from the points. When taking cards an adjusting stop limits the movement of the holder toward the shaft. The opening for the needle record is only 3¼ inches wide by 2½ inches high. In order that the needle record may be properly centered on the card a screw adjustment is provided to move the holder a short distance fore and aft parallel to the shaft axis. An adjustment to the holder is also provided so that both points will mark properly so as to prevent only one from recording. These two adjustments to the card holder will allow compensation for any expansion or warping that may occur.

Zero Reading.—If the meter were so set that both recording points described the same line on the card when there was no torsion in the shaft, it would be impossible to tell whether both points were or were not recording. For this reason it is customary to so adjust the points that they will record a small positive reading when the shaft is transmitting no power, that is, when the propellers are dragging but still

turning over. This positive reading is obtained by adjusting the conical set screws joining the butterfly piece to the radial arm yoke.

CALIBRATION.

To obtain the best results the shaft should be calibrated after the meter is installed in the section of shaft exactly as it is to be used on board ship, except for the card holder. The shaft is placed on two suitable supports and the end containing the fixed end of the meter is securely fastened to a fixed girder. The opposite end of the shaft is supported by a bearing which allows it to rotate or twist with as little friction as possible. To this free end of the shaft another girder is attached for applying the test loads. Whatever apparatus is used it is necessary that the turning forces due to the test loads applied at each end of the girder shall be balanced so that there will be no eccentric twist in the shaft. The usual method consists of applying known weights at one end of the girder and applying equal and opposite forces at the other end of the girder by means of a jack or turnbuckle. This jack or turnbuckle rests on a weighing scale for measuring the applied force. Suitable corrections are made for the weight pan and jack or turnbuckle attachments.

The weights in the pan at one end act downward while the turnbuckle must act with equal force in an upward direction at the other end of the girder. The points at which the turnbuckle and weight pan act upon the girder are knife edges that lie in a line passing through the shaft axis and are carefully measured therefrom. For convenience these points are usually taken equidistant from the shaft axis. Fig. 8 shows a general view of the shaft set up for calibration. The turnbuckle and scale are shown in the right foreground and the weight pan is shown in the right background. In Fig. 9 the weight pan is shown at the left and the turnbuckle and scale in the left background. The method of securing the fixed end of the shaft is shown in both figures.

The recording umbrella does not rotate around the shaft axis during this calibration test as it does when installed in

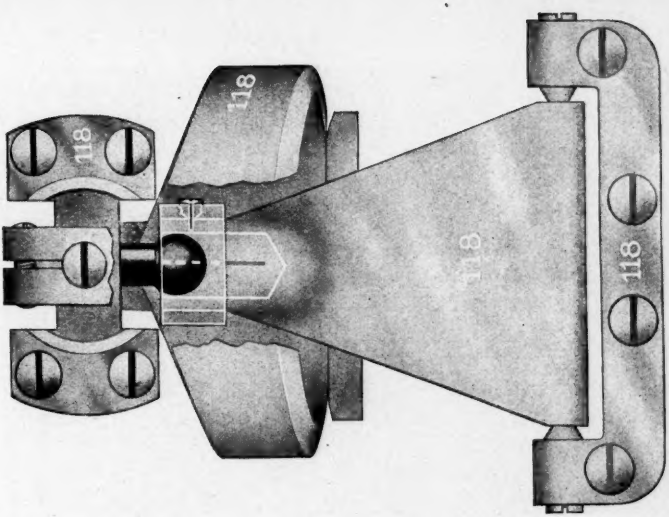


FIG. 6.—PHANTOM VIEW OF BUTTERFLY PIECE AND SLIDING BALL JOINT.

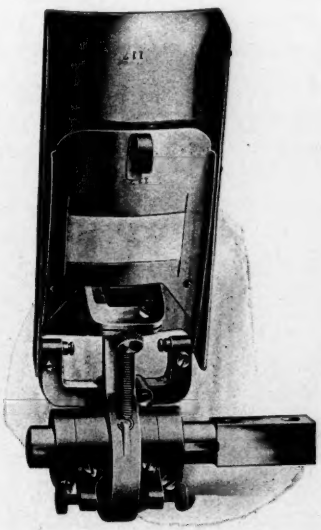


FIG. 7.—CARD-HOLDER. BY THE ADJUSTING SCREWS AT EITHER SIDE OF THE BOTTOM SUPPORT THE HOLDER CAN BE ROTATED UNTIL BOTH POINTS RECORD EVENLY.

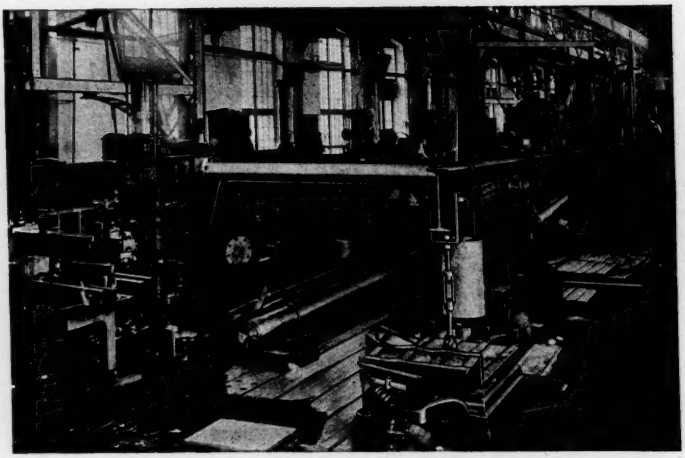


FIG. 8.—SHAFT SET UP FOR CALIBRATION.

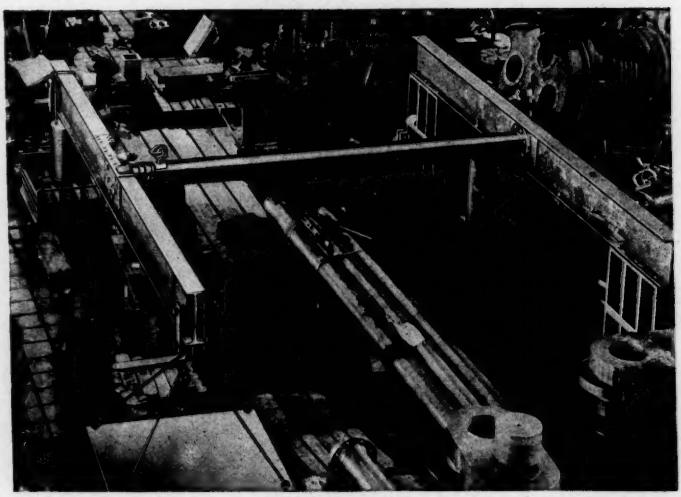


FIG. 9.—SHAFT SET UP FOR CALIBRATION. RECORDING UMBRELLA AND THE SPECIAL CARD-HOLDER SHOWN AT LEFT.

the ship. Provision must, therefore, be made for properly recording the angular twist obtained during this test. The record card is placed in a small holder carried by an arm that rests on a special plate on the shaft itself. This supporting plate is fitted to the circumference of the shaft, and when moved around the shaft causes the record card to describe a circle whose center is at the center line of the shaft. As the card passes the recording points two parallel lines are traced. The distance between these lines, corrected for the zero reading, is a measure of the moment or torque applied to the shaft.

The umbrella is mounted on the girder at the free end of the shaft. The method of mounting the umbrella and card holder is shown in Fig. 9.

Calibration Test.—1. Set up the shaft with the torsion meter installed as described above and as shown in Figs. 8 and 9.

2. Adjust the recording points so that the zero reading will be ten to fifteen-hundredths of an inch and take a card.

3. Apply a known weight to the weight pan and adjust the turnbuckle so an approximately equal upward thrust is exerted at the scale end and take another card.

4. Repeat operation 3 by adding a series of weights and, after adjusting the turnbuckle each time so the upward thrust is equal to the weights applied, take cards for each increment of weight. The shaft should be thoroughly rapped each time with lead hammers before readings are taken. These operations are continued until the desired load has been reached.

5. To check operation 4 similar cards are taken as the weights are removed step by step. From these results, with proper corrections for zero reading of the instrument, the calibration curve for the meter can easily be plotted. Torsion meter readings, corrected for zero reading, are plotted as abscissae and the torques applied to the shaft as ordinates. If the test is properly conducted the resulting curve will be a straight line, indicating that the angular twist in the shaft is directly proportional to the load applied. If the calibration curve is a straight line, it proves the correctness of the cali-

bration, provided no error, such as friction, has entered the calculation.

Torsion-Meter Constant.—From the calibration readings the torque per inch of meter reading can be calculated. Theoretically, this should be the same for all loads applied and, if the test is carefully conducted, will be practically so. Fig. 10 shows the calibration sheet for the shafts of the U. S. S. *Shaw*. By taking the average of torque per inch of meter readings for all loads a figure is obtained on which to base the shaft horsepower.

To facilitate ready calculation of the power transmitted the torsion meter constant contains, in addition to the torque per inch of meter reading, all other constants that appear in horsepower formula as follows:

Suppose

R = radius on which load is applied in feet
W = weight of load in pounds
R.P.M. = revolutions per minute
M = torque or moment in foot pounds
Q = foot pounds for 1 inch torsion meter reading
T = torsion-meter reading in inches;

then

$$\text{S.H.P.} = \frac{2 \times \pi \times R \times W \times \text{R.P.M.}}{33{,}000}$$

$$= \frac{2 \times \pi \times M \times \text{R.P.M.}}{33{,}000}$$

$$= \frac{2 \times \pi \times Q \times T \times \text{R.P.M.}}{33{,}000}.$$

Let

$$K = \text{torsion-meter constant} = \frac{2 \pi \times Q}{33{,}000},$$

then

$$\text{S.H.P.} = K \times T \times \text{R.P.M.}$$

					PORT SHAFT			STARBOARD SHAFT		
NUMBER	DOWNWARD PULL.	UPWARD THRUST	TOTAL PULL.	TOTAL TORQUE	METER READING	CORRECTED METER READING	TORQUE PER 1" METER READING	METER READING	CORRECTED METER READING	TORQUE PER 1" METER READING
1	0	0	0	0	.09	0		.04	0	
2	1000	1000	2000	19000	.48	.59	32203	.50	.56	33929
3	2000	2000	4000	38000	1.25	1.16	32759	1.17	1.13	33628
4	3000	3000	6000	57000	1.82	1.73	32948	1.74	1.70	33529
5	4000	4000	8000	76000	2.42	2.33	32618	2.32	2.28	33333
6	5000	5000	10000	95000	3.03	2.94	32313	2.89	2.85	33333
7	4000	4000	8000	76000	2.42	2.33	32618	2.30	2.26	33628
8	3000	3000	6000	57000	1.82	1.73	32948	1.74	1.70	33529
9	2000	2000	4000	38000	1.25	1.16	32759	1.16	1.12	33929
10	1000	1000	2000	19000	.67	.58	32759	.59	.55	34545
11	0	0	0	0	.09			.04	0	
			AVERAGE				32658			33709

M = MOMENT IN FOOT LBS.
S.H.P. = SHAFT HORSE POWER
Q = TORQUE FOR 1" TORSION METER READING
R.P.M. = REVOLUTIONS PER MINUTE
T = TORSION METER READING IN INCHES.
K = CONSTANT

$$S.H.P. = \frac{2\times P\times M\times R.P.M.}{33000} = \frac{2\times P\times Q}{33000}\times T\times R.P.M.$$

PORT SHAFT

$$S.H.P. = \frac{6.2832\times 32658\times T\times R.P.M.}{33000}$$

$$K = \frac{6.2832\times 32658}{33000}$$

$$K = 6.219$$

STARBOARD SHAFT

$$S.H.P. = \frac{6.2832\times 33709\times T\times R.P.M.}{33000}$$

$$K = \frac{6.2832\times 33709}{33000}$$

$$K = 6.418$$

$$S.H.P. = K\times T\times R.P.M.$$

FIG. 10.—CALIBRATION DATA, U. S. S. "SHAW."

From the above it is apparent that with the torsion-meter constant, torsion-meter reading, and corresponding revolutions per minute, the horsepower transmitted can be calculated and the card with the revolutions per minute entered thereon makes a permanent record for ship's files.

OPERATION ON BOARD SHIP.

Setting Up.—When the apparatus is out of use for any extended period, the recording apparatus is dismounted. To attach the apparatus first scrape all rust and dirt from the flange so that the umbrella bracket will fit tightly against it. Clean the top of the bar in the slot and the V-shaped block or butterfly piece that fits on to it. The butterfly piece should be adjusted by the centering screws so that it can be moved by the little finger. There must be no lost motion between this and its centers. In order to be on the safe side, make the centers a little tight. Adjust the umbrella so that it revolves freely in its centers without lost motion. Two dowels locate the bracket on the flange and it is secured by four machine screws.

Adjusting Recording Points.—A gage in the spare-part box fits the flange with a tongue projecting out to set the height of the recording points, Fig. 11. These are set with a watch key (number 4 key in the spare-part box, Fig. 12) so that they just touch the gage as it is moved around the flange. This brings both recording points at the same distance from the center of the shaft so both will record when the card is brought in contact. By means of the centering screws on the block attached to the top of the bar, set the points so that one is about $\frac{1}{8}$ inch farther from the flange than the other. The mark on the gage should be equidistant between the planes of the points. This is the approximate zero reading. Set the needles so that their planes of rotation do not cross when the shaft twists in ahead direction, otherwise the zero reading would have to be added to the reading taken when underway instead of being subtracted as it should be.

Adjust the centering screws so that the hinge in the umbrella piece was about as tight as it was before; then remove the umbrella bracket and make a final adjustment of the hinge screws. Do not disturb them too much or the zero reading will be thrown out. Put the umbrella bracket on the flange and secure. Try the umbrella for lost motion. If any is present, locate and remedy it. Try for tightness by insert-

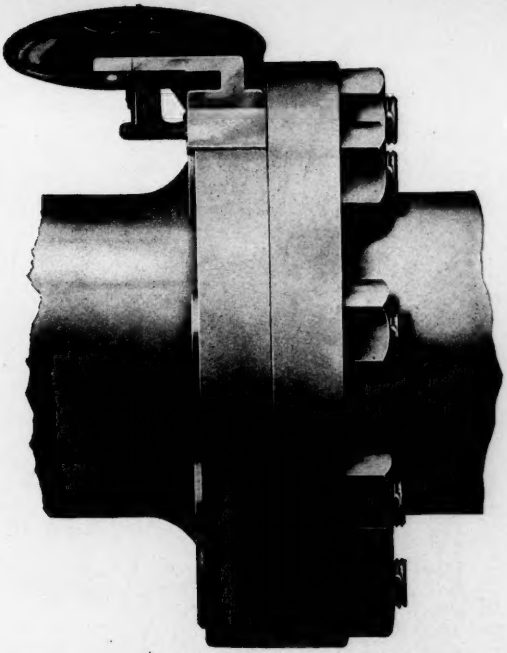


FIG. 11.—ADJUSTING RECORDING POINTS BY GAGE.

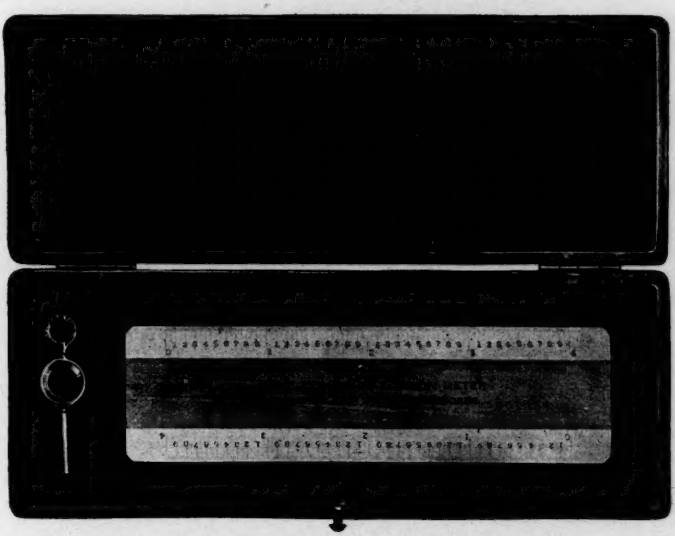


FIG. 12.—KEY FOR ADJUSTING RECORDING POINTS.
Scale for Measuring Cards (divided into inches and tenths).

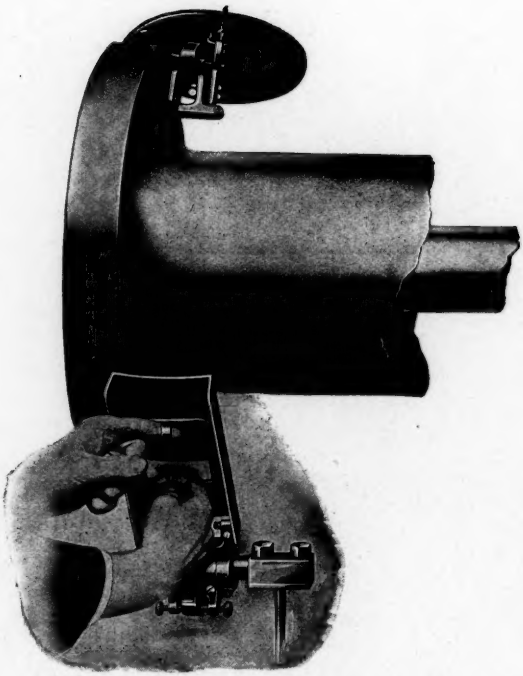


FIG. 13.—A CONVENIENT WAY OF TAKING CARDS.

ing a screw driver in the slot beside the bar and springing it. If the recording points do not come back to the original position when the strain is released, something is too tight, probably the centers of the umbrella or the sliding ball joint in the butterfly piece.

Sliding Ball Joint.—The later type meters have the sliding ball-joint end of the butterfly piece split with a collar containing an adjusting screw over the end, Fig. 5. If the ball fits loosely in this piece take up the adjusting screw. Lost motion here is not very serious, however, because when the shaft starts to revolve there is a slight force acting to bring the recording points towards zero. Therefore the lost motion would correct itself, as it is the difference between the readings when the shaft is dragging and when transmitting power that determines the horsepower. If the ball has become too worn the spare umbrella should be used. One is carried in the spare-part box of each ship equipped with these meters. In some cases the ball is not the same distance from the axis of the umbrella for all torsion meters on the same ship, in which case the spare umbrella is shipped without the ball attached. The spare balls are tagged for the shaft on which they may be used. In this event it is necessary to find the right ball and attach it to the spare umbrella or replace a worn ball on an umbrella already in use.

Adjusting the Card Holder.—Attach the card holder by two bolts to the support which is built out from the frame of the ship. Loosen the binding screws on the collars over the round bar, and move the holder forward or aft so that the center of the opening is in the plane of the axis of the umbrella as it revolves with the shaft. The object of this is to keep both points in the range of the card as the shaft twists. Jack the shaft until one point is directly opposite the opening. Turn the thumb stop screw to about one-half way and slacken the two slotted screws so that the holder may be revolved about the bar. With the card in the holder, push it until the point just touches the card. Release it, and the spring will bring the holder back away from the points. Adjust the

screws and move the thumb screw until the stop is properly set to keep the card away from the points. This prevents the points from striking the metal when the first reading is taken after the ship is underway. Tighten up all screws.

To Determine Zero Readings.—The zero reading is obtained by dragging the shaft. This is done by getting some way on the ship, about one-third speed, then shutting off the steam to the turbine, including auxiliary exhaust, and permitting the shafts to revolve partly by the ship's and partly by its own inertia and that of the rotor. Cards are taken at frequent intervals while the shaft gradually slows down and the minimum of these readings is taken as the zero reading. As the shaft soon stops revolving after steam is shut off, it is advisable to take a few practice cards before the drag actually starts. One man, experienced at taking cards, should be at each torsion meter while the shafts are dragging.

Taking Cards.—Since the stop is habitually set so that the points will not touch the cards when the ship gets underway, the first thing to be done is to so adjust this stop that the points just make a mark without tearing the card. About a dozen cards should be placed in the holder. Some prefer to adjust the stop so that the needles are about $\frac{1}{16}$ inch away from the card; then, with a slight pressure of the forefinger placed horizontally behind the card, bulge it outward slightly, just enough to make the points mark, as shown in Fig. 13. The holder is sprung forward with the other hand at the same time. This method is used by all experienced operators because it lessens the danger of the points striking metal and does not require such fine adjustment of the stop screw. It is attended, however, with some danger of cutting the fingers by the revolving points and should be attempted only by careful operators. Should one point mark deeper than the other, do not push the card farther in; that will only cause the first point to cut. Instead, adjust the side thumb screws revolving the card holder about its center, backing off one and setting up on the other. (Be sure the holder is tightly held between both adjusting screws before taking another record.) Repeat this until both points mark evenly.

Calculating the Horsepower.—Take a card and at the same time determine the revolutions per minute. Note the revolutions on the card. Also, record on the card from which shaft it is taken, the date and time. The linear distance in inches between the two parallel marks on the cards, minus the zero reading (in rare cases this is plus if the planes of points cross as the shaft twists) is the correct torsion-meter reading. A convenient scale is furnished each ship for measuring cards, Fig. 12.

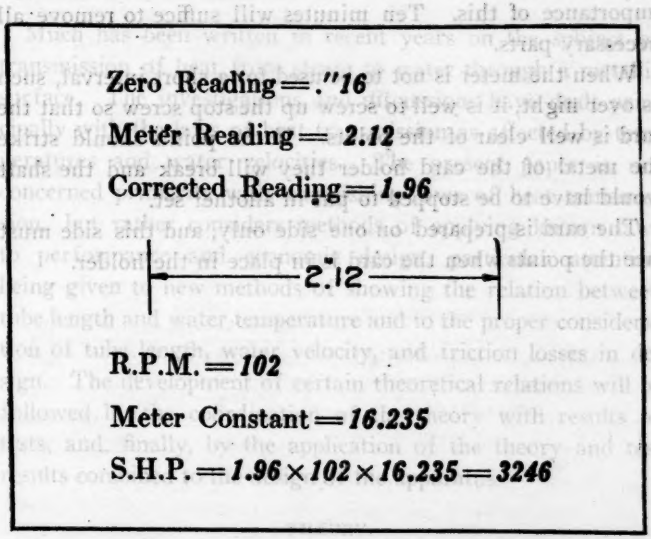


FIG. 14.—SHOWS A CARD AS TAKEN BY THE TORSION METER, TOGETHER WITH THE NECESSARY CALCULATION TO OBTAIN THE SHAFT HORSEPOWER.

Multiply the corrected reading (in inches and fractions thereof) by the torsion-meter constant and this result by the revolutions per minute at the time the card was taken. The final result is the horsepower being transmitted through the shaft. The torsion-meter constant is determined from shop calibration of the shaft, as already described, and this constant for each shaft is furnished by the shipbuilder to each ship.

Care of Instrument.—The twist of a large shaft as recorded by the torsion meter is sometimes as small as one ten-thousandth of an inch. To record such minute angles accurately the recording parts are made light and delicate and the instrument must be carefully installed and operated.

When meters are not to be used for some time, remove and stow away all parts on the outside of the shaft and the card holder until needed to prevent their becoming dirty and possibly injured. Too much emphasis cannot be laid upon the importance of this. Ten minutes will suffice to remove all necessary parts.

When the meter is not to be used for a short interval, such as over night, it is well to screw up the stop screw so that the card is well clear of the points. If the points should strike the metal of the card holder they will break and the shaft would have to be stopped to put in another set.

The card is prepared on one side only, and this side must face the points when the card is in place in the holder.

PERFORMANCE AND DESIGN OF SURFACE FEED-WATER HEATERS.

By M. C. Stuart, Mechanical Engineer, Associate.

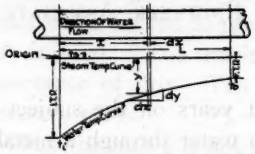
Much has been written in recent years on the subject of transmission of heat from steam to water through a metallic surface. The investigations and discussions have dealt principally with the laws of heat transmission as affected by temperatures and water velocities. The present paper is not concerned primarily with establishing laws of heat transmission, but rather considers methods of applying known laws to performance and economic design, particular attention being given to new methods of showing the relation between tube length and water temperature and to the proper consideration of tube length, water velocity, and friction losses in design. The development of certain theoretical relations will be followed by the coördination of the theory with results of tests, and, finally, by the application of the theory and test results combined to the design of the apparatus.

THEORY.

The theory of heat transmission as developed herein is based upon the fundamental law, credited to various investigators since the time of Newton, that heat transmission is proportional to temperature difference. Referring to the diagram in Fig. 1, which represents the elementary case of water flowing through a tube which is surrounded by steam, it is evident that with a constant water velocity and a constant steam pressure, the temperature of the water will increase from inlet to outlet along some curve such as that marked "Water Temperature Curve." The method of treatment will

consist of first deriving the equation of this curve of temperature rise of the water, then determining the mean temperature difference between the steam and water, which is the mean ordinate of the curve, and, finally, determining the heat transfer factor.

The assumption that the amount of heat transferred at any



I Derivation Of Temperature Rise Curve.

L.T.D = Initial Temperature Difference, At Beginning Of Length "L".
F.T.D = Final Temperature Difference, At Distance "L" Or Unit Distance.
y = Temperature Difference At Distance "x" From Cold End.

$$\frac{dy}{dx} = -by \; ; \; \frac{dy}{y} = -b\,dx$$

Integrating We Obtain: $\log_e y = -bx + C_1$, Whence $y = ae^{-bx}$ ①

When $x = 0$, $y = L.T.D.$ Substituting In ① $L.T.D. = ae^0 = a$ ∴ $y = L.T.D.\,e^{-bx}$ ②

When $x = L$ Or Unit Length, $y = F.T.D.$ Substituting In ② We Obtain:

$$F.T.D. = L.T.D.\,e^{-b} \quad \text{or} \quad e^{-b} = \frac{F.T.D.}{L.T.D.}$$

$$e^{b} = \frac{L.T.D.}{F.T.D.} \quad b\log_e e = \log_e \frac{L.T.D.}{F.T.D.} \quad b = \log_e \frac{L.T.D.}{F.T.D.}$$

Substituting Values Of "a" And "b" In Equation ① , We Obtain:

$$y = \frac{L.T.D.}{e^{x \log_e \frac{L.T.D.}{F.T.D.}}}$$ To Simplify Denominator.

Let $e^{x \log_e \frac{L.T.D.}{F.T.D.}} = z$

$$\log_e z = x \log_e \frac{L.T.D.}{F.T.D.} \quad x \log_e e = x \log_e \frac{L.T.D.}{F.T.D.} \quad \text{Whence } z = \left(\frac{L.T.D.}{F.T.D.}\right)^x$$

Substituting This Value Of z In Equation $y = \frac{L.T.D.}{z}$, We Obtain:

$$y = \frac{L.T.D.}{\left(\frac{L.T.D.}{F.T.D.}\right)^x} \quad ⑤$$

y = F.T.D. At End Of Length x = $(F.T.D.)_x$; $(L.T.D.)_x$ = Temperature Difference At Beginning Of Length "x"

$$(F.T.D.)_x = \frac{(L.T.D.)_x}{\left(\frac{L.T.D.}{F.T.D.}\right)^x} \quad ④$$

FIG. 1.—DERIVATION OF CURVE OF TEMPERATURE RISE.

II Mean Temperature Difference Between Water & Steam.

T_m = Mean Temperature Difference Between Water And Steam
A = Area Between Curves Of Water And Steam Temperature.
t_s = Steam Temperature, Degrees, Fahr
t_1 = Inlet Water Temperature, Degrees, Fahr.
t_0 = Outlet Water Temperature, Degrees, Fahr
L = Unit Length Of Tube.

$$A = \int_0^L y\,dx \; , \; T_m = \frac{A}{L} = \frac{1}{L}\int_0^L y\,dx = \int_0^1 y\,dx$$

From Equation ⑤ $y = \frac{t_s - t_1}{\left(\frac{t_s - t_1}{t_s - t_0}\right)^x}$

$$T_m = \int_0^1 \frac{t_s - t_1}{\left(\frac{t_s - t_1}{t_s - t_0}\right)^x} dx = -(t_s - t_1)\int_0^1 \left(\frac{t_s - t_1}{t_s - t_0}\right)^{-x} dx = -(t_s - t_1)\left[\frac{\left(\frac{t_s - t_1}{t_s - t_0}\right)^{-x}}{\log_e \frac{t_s - t_1}{t_s - t_0}}\right]_0^1$$

$$T_m = \frac{t_s - t_1}{\log_e \frac{t_s - t_1}{t_s - t_0}} - \frac{t_s - t_0}{\log_e \frac{t_s - t_1}{t_s - t_0}} = \frac{(t_s - t_1) - (t_s - t_0)}{\log_e \frac{t_s - t_1}{t_s - t_0}} \quad ③$$

$$T_m = \frac{t_0 - t_1}{\log_e \frac{t_s - t_1}{t_s - t_0}} = \frac{\text{Temp. Rise}}{\log_e \frac{(T.D.)}{(F.T.D.)}} \quad ⑥$$

III Heat Transfer Rate, Steam To Water.

K = Heat Transfer Rate, B.t.u. Per Hr., Per Sq. Ft. Of Heating Surface, Per Degree Mean Temperature Difference.
W = Weight Of Water, Pounds Per Hour.
S = Heating Surface, Square Feet.

From Definition $K = \frac{W(t_0 - t_1)}{S \times T_m}$ ⑦

Substituting Value Of T_m From Equation ⑥ We Obtain:

$$K = \frac{W(t_0 - t_1)}{S \times \frac{(t_0 - t_1)}{\log_e \frac{(T.D.)}{(F.T.D.)}}} = \frac{W \log_e \frac{(T.D.)}{(F.T.D.)}}{S} = \frac{W \log_e \frac{(T.D.)}{(F.T.D.)}}{S} \quad ⑧$$

FIG. 2.—DERIVATION OF CURVE OF MEAN TEMPERATURE DIFFERENCE AND HEAT TRANSFER COEFFICIENT.

point along the tube is proportional to the temperature difference between the steam and water at that point may be expressed by the differential equation: $\frac{dy}{dx} = -by$ which, after integration, reduces to the equation of the temperature rise curve, $y = ae^{-bx}$ ... (1), as given in Fig. 1. After determining the values of the constants a and b in terms of temperatures, this equation simplifies to the form of equations (3) or (4), given in Fig. 1.

The mean temperature difference may be considered as the average distance between the curves of steam temperature and water temperature. In Fig. 2 the value of the mean temperature difference is derived by the process of determining the mean ordinate of the temperature-rise curve by dividing the area between the two curves by the length. This derivation differs from the usual method of writing a heat balance, which involves quantity of water, amount of surface, and the heat transfer factor. The result is the usual forms for mean temperature difference as given in equations (5) and (6), Fig. 2.

Having determined the mean temperature difference, the heat transfer factor, K, may be written directly from its definition, K = B. t. u., transferred per hour, per square foot of surface, per degree mean temperature difference. This results in the usual form for heat transfer factor as given in equation (7), Fig. 2, or the somewhat simpler form of equation (8).

PERFORMANCE OF HEATERS; TEMPERATURE DIAGRAM.

The validity of the theoretical relations between temperatures and tube length may be most clearly demonstrated from tests of multi-pass feed-water heaters, in which temperature readings are obtained at the end of each pass, because such tests give simultaneous data for performance of identical sections of a heater with identical conditions of steam pressure and water flow. A recent test of a four-pass heater, made at the U. S. Naval Engineering Experiment Station, Annapolis, Md., gave the following data for a run in which the water velocity was 5.14 feet per second and the steam temperature was 227.2 degrees F.

WATER TEMPERATURES, DEGREES F.

	Inlet.	Outlet.
First pass	85.6	133.4
Second pass	133.4	165.8
Third pass	165.8	186.7
Fourth pass	186.7	201.3

The writer plots these temperatures in Fig. 3, in what may be termed a "temperature diagram." The abscissae are the inlet temperatures of the several passes and the ordinates are the corresponding outlet temperatures. A base line is drawn diagonally across the diagram through points of equal temperature, and the line of inlet temperature *vs.* outlet temperature, which is drawn through the plotted points, is extended to meet the base line at a point corresponding to the steam

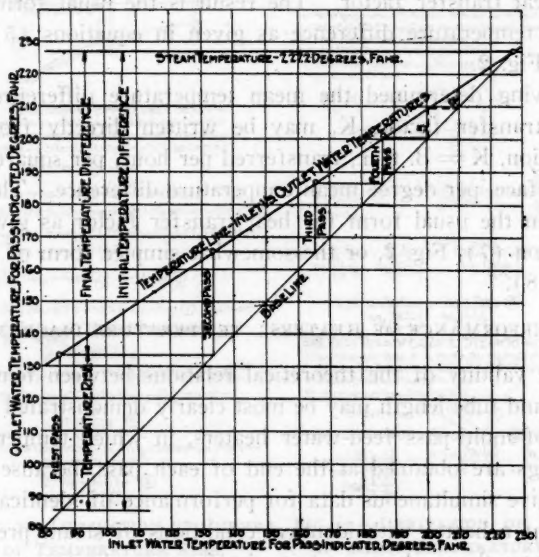


FIG. 3.—TEMPERATURE DIAGRAM FOR FOUR-PASS FEED-WATER HEATER, BY SEPARATE PASSES.

temperature, from the following consideration. If water enters any pass at the temperature of the steam, there will, of course, be no temperature rise, and the temperatures of inlet water, outlet water and steam will all be equal. Therefore the line of inlet *vs.* outlet temperatures intersects the base line at the steam temperature. In a multi-pass heater, the outlet temperature of one pass becomes the inlet temperature of the next pass. This feature may be shown graphically by the

"stairway" formed between the temperature line and the base line. The temperatures to be expected by adding additional passes in series may be obtained by extending the stairway, as shown by the dotted lines for 5th and 6th passes.

A relation between the temperature diagram and the theory as previously developed will now be pointed out. If the temperature line on the diagram is straight and intersects the base line at the steam temperature, it is evident that the ratio of

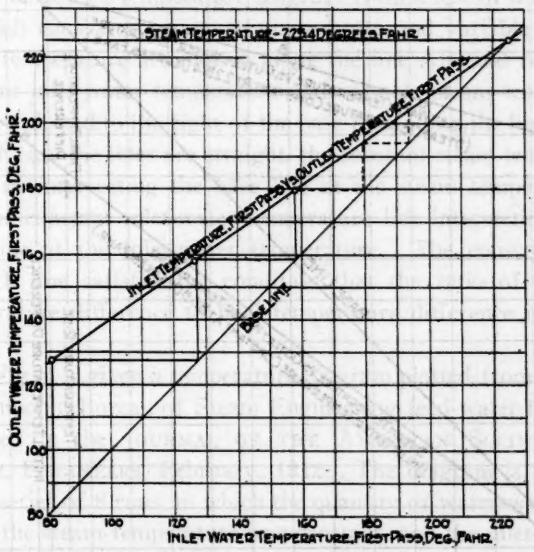


FIG. 4.—TEMPERATURE DIAGRAM FOR FIRST PASS OF MULTI-PASS HEATER.

initial temperature difference to final temperature difference is constant. From formula (8) for the value of the heat transfer coefficient, it is seen that for constant values of W and S, as exist in the several passes of the multi-pass heater, K will be constant if the ratio (I.T.D.) to (F.T.D.) is constant. Inasmuch as the temperature diagram satisfies this condition, that part of the theory is borne out which states that the heat transfer coefficient is independent of temperatures, and de-

pends only upon the ratio of initial to final temperature differences.

The principle of the temperature diagram, which has previously been discussed by the writer in connection with lubricating-oil coolers,* has a wide application in the analysis of performance of heat transfer apparatus, and will be illustrated and discussed briefly in its several applications. The diagram

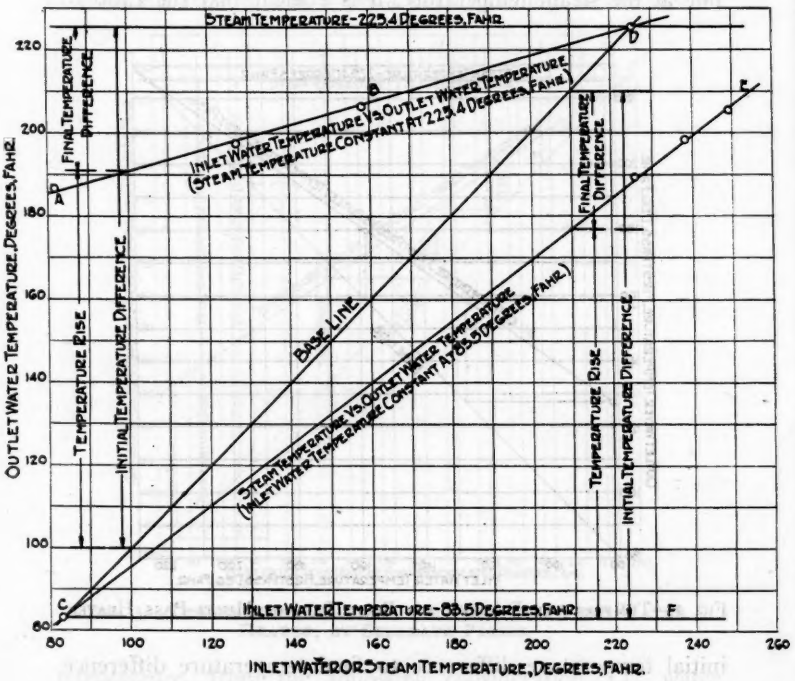


FIG. 5.—TEMPERATURE DIAGRAM FOR HEATER, SHOWING VARIABLE STEAM AND WATER TEMPERATURES.

of Fig. 3. was for the separate passes of a four-pass heater. In Fig. 4 is given a diagram for the first pass only, plotted from three separate runs in which the inlet temperature for

* "The Performance of Lubricating-Oil Coolers," JOURNAL A. S. N. E., May, 1917, page 300.

one run was adjusted to equal the outlet temperature of the previous run. This diagram for a single pass has the same characteristics as the diagram for a number of similar passes arranged in series. The temperature line is extended to meet the base line at the steam temperature, and the stairway between the temperature line and the base line shows the temperatures which would be obtained by arranging a number of similar passes in series.

Fig. 5 shows a temperature diagram from a test of a heater in which runs at constant steam pressure and variable inlet-water temperature are plotted along the line AB, and runs at constant inlet-water temperature and variable steam temperature are plotted to the right of the base line, along the line CE. In each case the lines are straight, the constant steam-temperature line intersecting the base line at the steam temperature and the constant inlet-water-temperature line intersecting the base line at the inlet-water temperature. The construction of each line satisfies the condition that the ratio of initial temperature difference to final temperature difference is constant.

In Fig. 6 is given a temperature diagram plotted from a test made upon a Bureau of Steam Engineering feed-water heater, reported in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, February, 1912. The diagram is drawn for a series of 8 runs, in which the quantity of water was constant, the steam temperature was constant, and the inlet-water temperature was varied. The line of temperatures of inlet water *vs.* outlet water is practically straight and intersects the base line at the steam temperature. The water temperatures which would result by placing any number of similar heaters in series may be obtained by constructing the stairway between the temperature line and the base line. For example, starting with an inlet temperature of 90 degrees F at the first heater, a temperature of 207 degrees would be obtained at the outlet of the eighth heater in series. The curve of temperature rise throughout the series of heaters is obtained by pro-

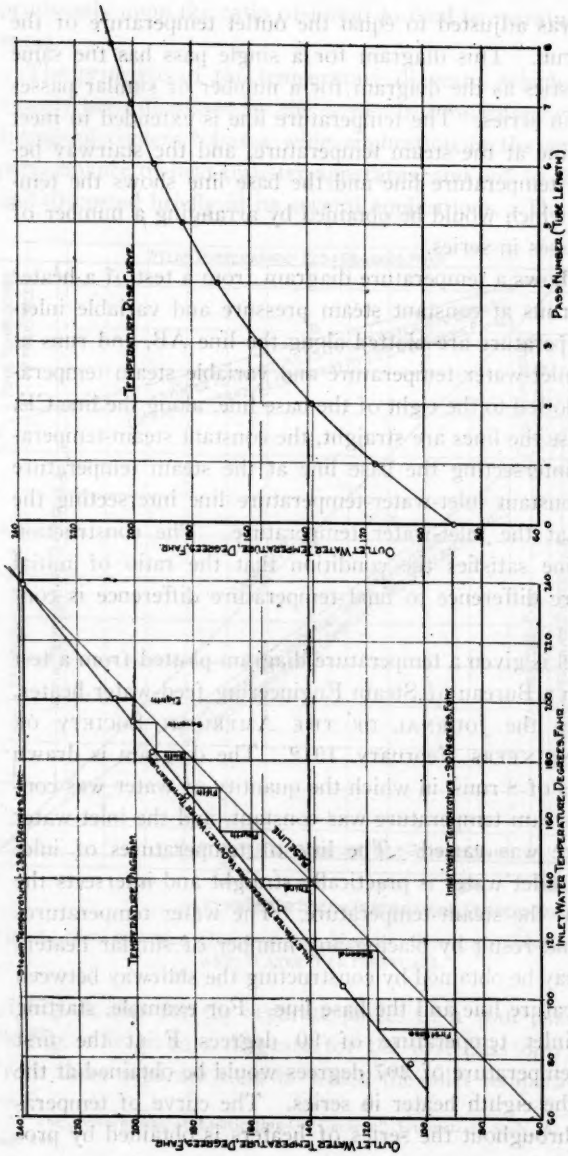

Temperature Rise Curve.
Pass Number (Tube Length)
Outlet Water Temperature, Degrees Fahr.
Temperature Diagram.
Inlet Water Temperature, Degrees Fahr.
Test of Bureau Heater
Reported in the Journal of the American Society of Naval Engineers, February 1912.
Runs No's 8-15, Pp 155-166 Used in Plotting.
Tested by O. Z. Howard.
Weight of Water-175,160 Pounds per Hour.

FIG. 6.—TEMPERATURE DIAGRAM FOR BUREAU HEATER.

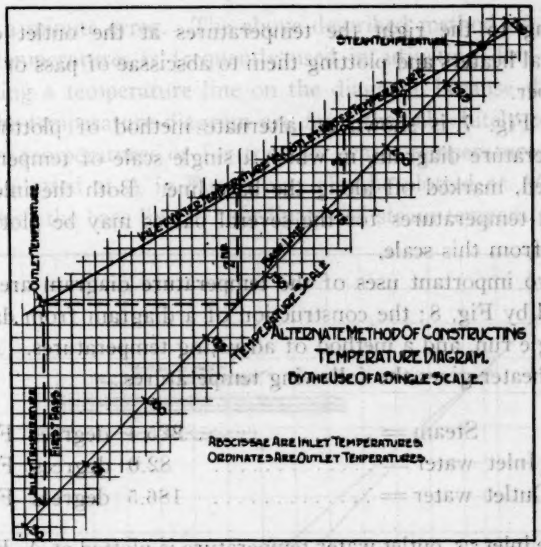

STEAM TEMPERATURE
OUTLET TEMPERATURE
ALTERNATE METHOD OF CONSTRUCTING TEMPERATURE DIAGRAM.
BY THE USE OF A SINGLE SCALE.
ABSCISSAE ARE INLET TEMPERATURES.
ORDINATES ARE OUTLET TEMPERATURES.

FIG. 7.—ALTERNATE METHOD OF CONSTRUCTING TEMPERATURE DIAGRAM.

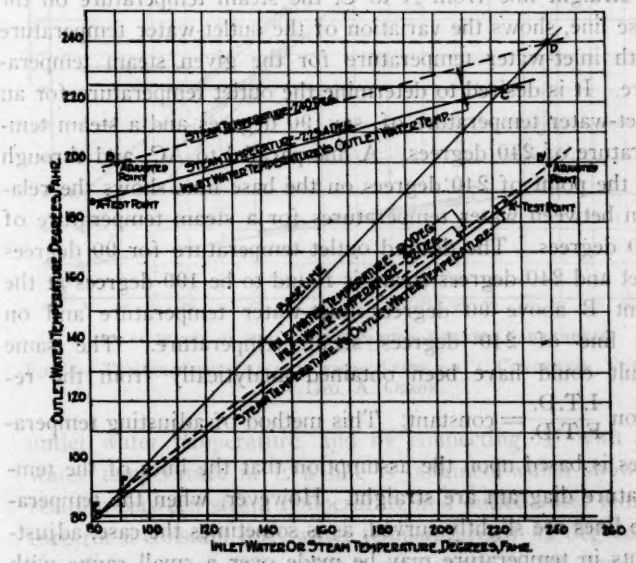


FIG. 8.—USE OF TEMPERATURE DIAGRAM IN MAKING TEMPERATURE ADJUSTMENTS.

jecting to the right the temperatures at the outlet of the several heaters and plotting them to abscissae of pass or heater number.

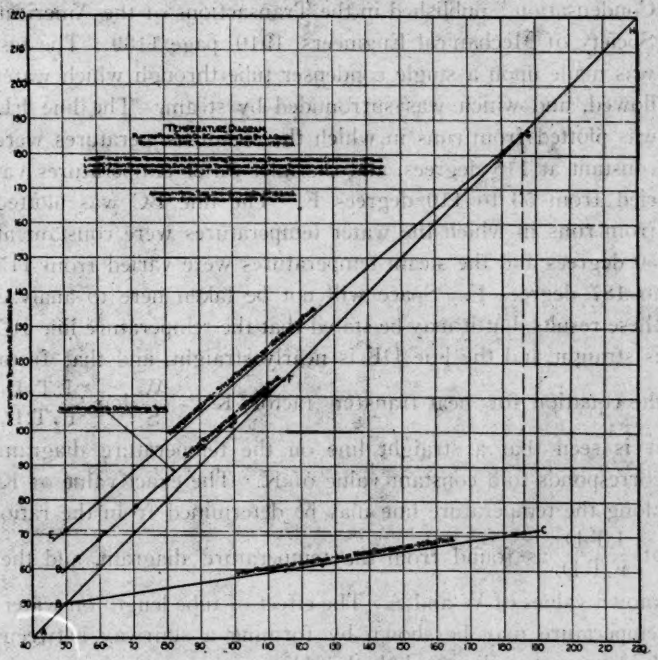
In Fig. 7 is shown an alternate method of plotting the temperature diagram, in which a single scale of temperatures is used, marked off along the base line. Both the inlet and outlet temperatures for the several passes may be plotted or read from this scale.

Two important uses of the temperature diagram are illustrated by Fig. 8; the construction of a diagram from data of a single run, and a method of adjusting temperatures. A test on a heater gives the following temperatures:

Steam = ...	225.4	degrees F.
Inlet water = ...	82.0	degrees F.
Outlet water = ...	186.5	degrees F.

The inlet *vs.* outlet water temperature is plotted at A, Fig. 8. A straight line from A to C, the steam temperature on the base line, shows the variation of the outlet-water temperature with inlet-water temperature for the given steam temperature. It is desired to determine the outlet temperature for an inlet-water temperature of, say, 90 degrees and a steam temperature of 240 degrees. A line parallel to AC, and through D, the point of 240 degrees on the base line, shows the relation between water temperatures for a steam temperature of 240 degrees. The desired outlet temperature for 90 degrees inlet and 240 degrees steam is found to be 199 degrees at the point B above 90 degrees inlet-water temperature and on the line of 240 degrees steam temperature. The same result could have been obtained analytically from the relation $\frac{\text{I.T.D.}}{\text{F.T.D.}} = \text{constant}$. This method of adjusting temperatures is based upon the assumption that the lines of the temperature diagram are straight. However, when the temperature lines are slightly curved, as is sometimes the case, adjustments in temperature may be made over a small range with

only a minute error. The above described method of adjusting temperatures is frequently used in adjusting data before plotting a temperature line on the diagram, because the lines on the temperature diagram are for either absolutely constant steam temperatures or for constant water temperatures. The data plotted at A in Fig. 8 may also be plotted at A′ to the right of the base line to coördinates of steam temperatures *vs.*



INLET WATER OR STEAM TEMPERATURE, DEGREES FAHRENHEIT.

FIG. 9.—TEMPERATURE DIAGRAM FROM TESTS ON A CONDENSER TUBE, BY GEO. A. ORROK.

outlet water temperature, and by connecting A′ with inlet-water temperature at E a line is obtained which shows the relation between steam temperature and outlet-water temperature for a constant inlet-water temperature of 82 degrees. A line for constant inlet-water temperature of 90 degrees is

drawn parallel to A'E, through 90 degrees on the base line. By drawing other parallel lines a complete temperature diagram may be constructed showing the outlet temperature for any desired inlet or steam temperatures.

To extend the method to another field, there is shown in Fig. 9 a temperature diagram plotted from the classic test of Mr. G. A. Orrok on "The Transmission of Heat in Surface Condensation," published in the Transactions of the American Society of Mechanical Engineers, 1910, page 1139. The test was made upon a single condenser tube through which water flowed, and which was surrounded by steam. The line FD was plotted from runs in which the steam temperatures were constant at 115 degrees, and the inlet-water temperatures varied from 53 to 110 degrees F. The line BC was plotted from runs in which the water temperatures were constant at 50 degrees and the steam temperatures were varied from 115 to 187 degrees F. Space will not be taken here to analyze these results, but it may be stated that the temperature line BC is straight and the line DF is nearly straight, and that from the equation for heat transfer factor $K = \frac{W}{S} \log_e \frac{I.T.D.}{F.T.D.}$ it is seen that a straight line on the temperature diagram corresponds to a constant value of K. The exact value of K along the temperature line may be determined from the ratio of $\frac{I.T.D.}{F.T.D.}$ as found from the temperature diagram, and the known values of W and S. The effect of tube length on water temperature may be shown by forming a stairway between the temperature line and the base line.

One other example of the temperature diagram is given in Fig. 10, in this case the data being obtained from the experiments of the Babcock & Wilcox Co., on the transfer of heat from a hot gas to water through a boiler tube.* The apparatus consisted of a 20-foot copper pipe of 2 inches internal

* "Experiments on the Rate of Heat Transfer from a Hot Gas to a Cooler Metallic Surface," Babcock & Wilcox Co., New York, 1916, page 56, Run No. 3.

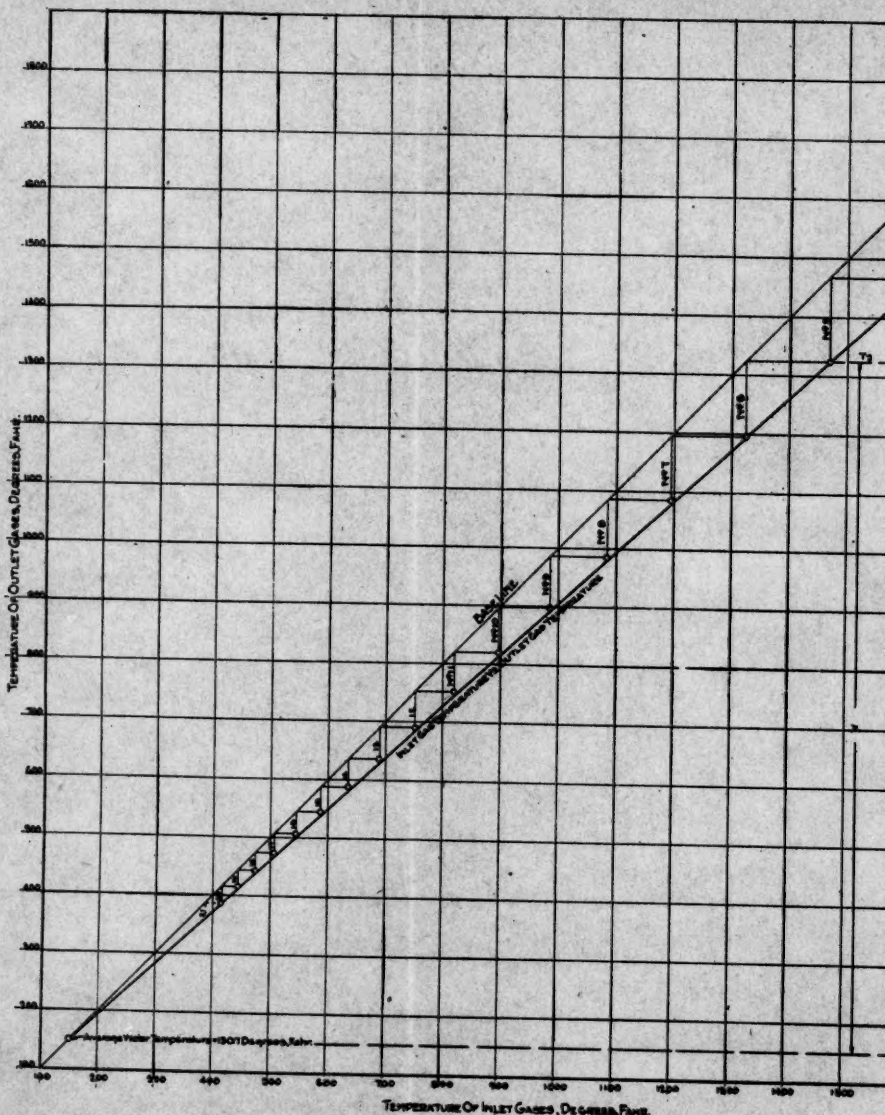
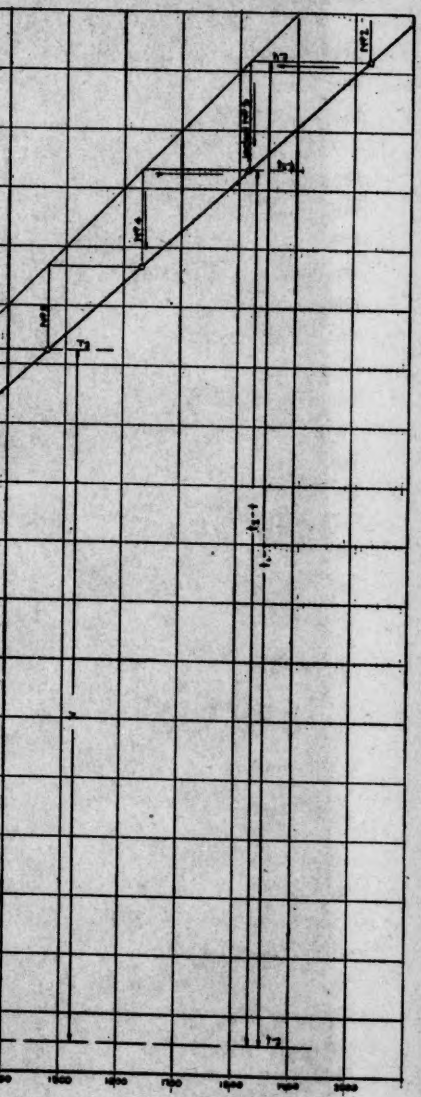


FIG. 10.—TEMPERATURE DIAGRAM FROM TESTS ON A BOILER TUBE BY THE



BY THE BABCOCK & WILCOX CO.

diameter, surrounded by 20 individual water jackets. Heated gas passed through the pipe, and the temperatures at the inlet and outlet of each foot-length section were computed from the amount of heat removed by the water circulated in the jackets and the specific heat of the gases. The inlet and outlet temperatures of the gases for the 20 sections are plotted in Fig. 10 in the form of a temperature diagram whose abscissae are inlet temperatures of gases and ordinates are outlet temperatures of gases. A straight line through the points intersects the base line at the water temperature. Points 2 to 6, inclusive, fall exactly on the line, and points 7 to 19 are removed from the line by a constant amount of about 5 degrees F. The points are connected by the stairway which when extended at either end of the diagram would give the relation between additional tube length and temperatures. In the case of gases, a straight line on the diagram does not mean a constant value of heat transfer coefficient on account of the variable value of specific heat of the gases. However, it appears that the temperature diagram would be quite useful in analyzing results of tests of this kind.

TUBE LENGTH AND TEMPERATURE RISE.

The analytical relation between tube length and temperature rise along the tube was derived in Fig. 1, and is represented by formula (4). In Fig. 11 are given three graphical methods of showing the relation between tube length and water temperatures. The first method is the temperature diagram, where, by means of a stairway formed between the temperature line and the base line, the temperatures for any number of equal lengths may be determined. By projecting the temperatures from the stairway of the temperature diagram to the right, there is obtained the temperature-rise curve to coördinates of tube length and temperature. A third method is derived by taking the logarithm of both sides of the equation of the curve of temperature rise:

$$y = ae^{-bx}.$$

This produces the equation $\log_e y = \log_e a^{-bx}$, which may be plotted as a straight line of semi-logarithmic coördinates, *i. e.*, arithmetical abscissae of tube length and logarithmic ordinates of temperature differences between steam and water. Data from a test of a four-pass heater are plotted on

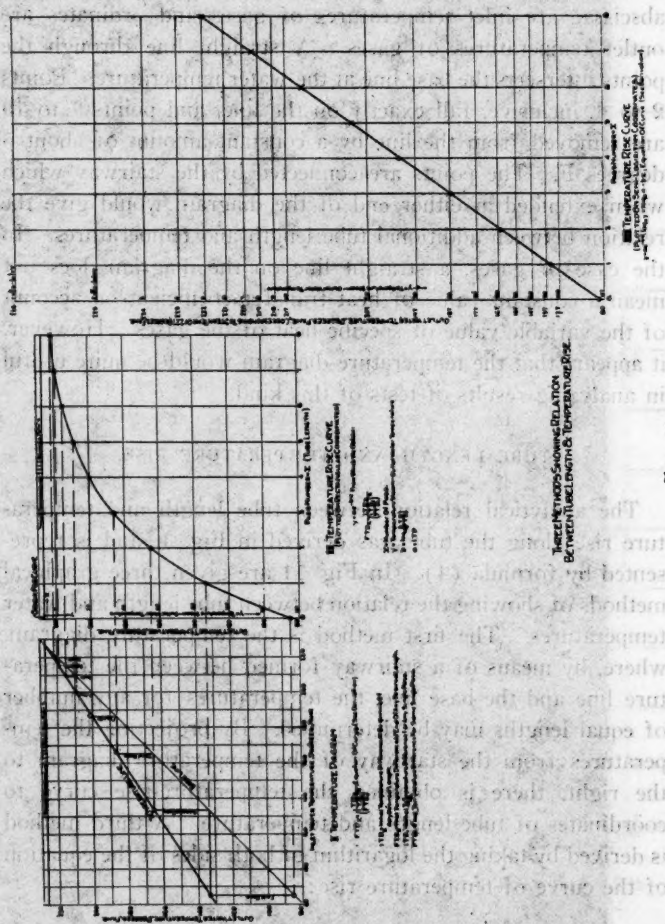


FIG. 11.

the curves for each of these three methods in Fig. 11. It is noted that in the temperature-rise curve the origin is at the upper left-hand corner, and that the ordinates, measured downward from the origin, are temperature differences between steam and water. The corresponding water temperatures are marked along the ordinates for convenience. In the semi-log plot, the ordinate starts with a value of $(t_s - t_o) = 1$ and the absolute origin, where $(t_s - t_o) = 0$, is at infinity. Either of these graphical methods, or the equation of the curve, may be used to solve problems of temperature rise as affected by tube length for a constant velocity. If it is desired to determine the temperatures for a tube length which is an even multiple of a tube length for which the temperatures are known, the simplest method is the use of the stairway on the temperature diagram.

For uneven multiples of lengths the semi-log. curve is more convenient. This method is also more accurate for temperatures near the steam temperature, on account of the spacing of the temperature ordinates. In order to draw the semi-log. curve it is only necessary to know the water temperatures at the beginning and end of a known tube length and the steam temperature. Using tube length as abcissae and logarithms of $(t_s - t_o)$ as ordinates, the inlet- and outlet-water temperatures are plotted and a straight line drawn through the points.

TEST OF MULTI-PASS HEATER.

In order to extend the discussion to cover various water velocities, and to apply the methods to the performance and design of a heater, a test of an 8-pass feed-water heater will be cited. The principal dimensions of the heater were:

Number of tubes	88
Number of passes	8
Number of tubes per pass	11
Length of pass, between tube sheets, feet and inches	3–0½
Length of unit tube, feet and inches	30–4
Total length of tubing in heater, feet and inches	333–8
Heating surface, outside of tubes, square feet	54.6
Outside diameter of tubes, inches	0.625
Thickness of tubes, B.W.G., No	20

The heater was installed with the tubes in a vertical position. Steam was supplied the heater at a pressure of 10 pounds gage. The inlet-water temperature was approximately 90 degrees F. Readings of water temperature were taken at the heater inlet and at the outlet of the 2d, 4th, 6th and 8th passes. The condensed steam was collected in a drain pot in which a

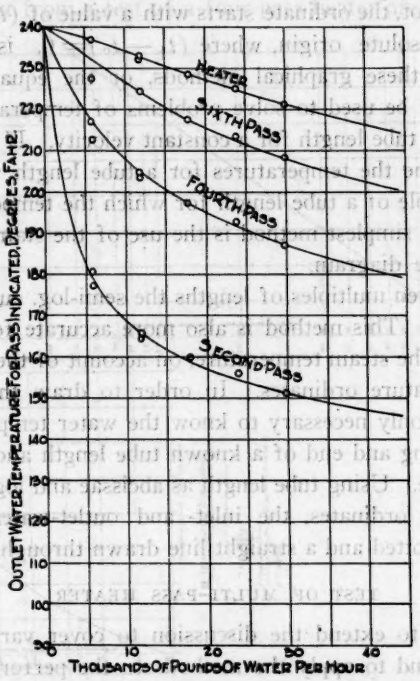


FIG. 12.—PERFORMANCE CURVES FOR AN EIGHT-PASS HEATER.

water level was maintained. Air was relieved from the steam space through air cocks. The quantities of water and steam were obtained by weighing. Analysis of the comparative performance of the several passes were made by plotting the inlet and outlet temperatures in temperature diagrams similar to that of Fig. 7. In nearly all cases the temperature lines were straight. The outlet temperatures were adjusted to correspond

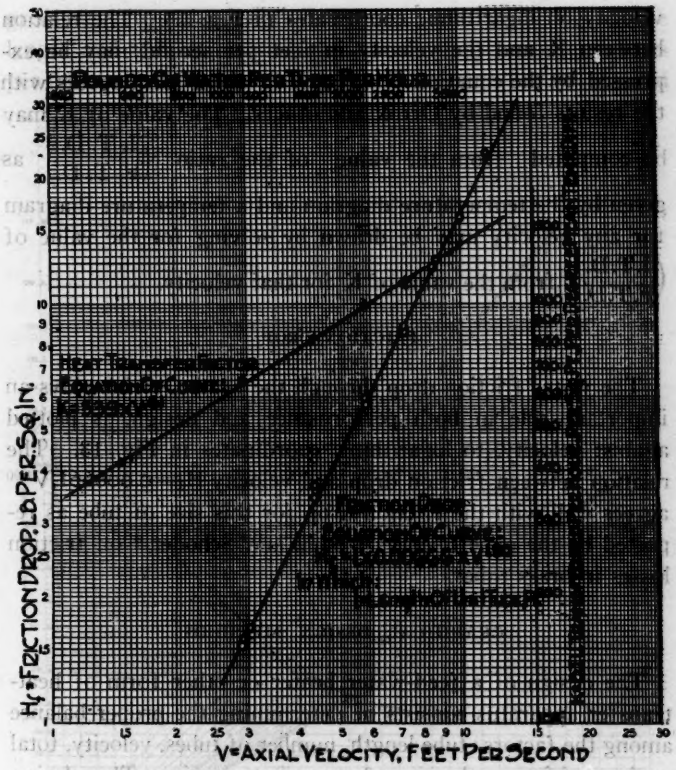


FIG. 13.—LOGARITHMIC PLOTS OF HEAT TRANSFER FACTORS AND FRICTION DROP.

to a steam temperature of exactly 240 degrees and an inlet-water temperature of 90 degrees F. by the method shown in Fig. 8. The adjusted temperatures were then plotted in Fig. 12 to coördinates of capacity and temperatures at the outlet of the several passes.

HEAT TRANSFER FACTOR.

The values of heat transfer coefficients, as computed from the formula $K = \frac{W}{S} \log_e \left(\frac{I.T.D.}{F.T.D.} \right)$ are plotted against water

velocity on logarithmic coördinates in Fig. 13. The relation between K and the velocity in feet per second may be expressed by the equation $K = 339\ V^{.61}$, which form agrees with the results found by Orrok and others. The value of K may be computed from the value of the ratio $\left(\frac{\text{I. T. D.}}{\text{F. T. D.}}\right)$ as given by the temperature diagram and a temperature diagram for any velocity may be drawn by solving for the value of $\left(\frac{\text{I.T.D.}}{\text{F.T.D.}}\right)$ from the value of K for that velocity.

FRICTION DROP.

The water friction drop through the heater, which plays an important rôle in both performance and design, is plotted against velocity, to logarithmic coördinates in Fig. 13. The relation between friction drop and velocity $H_f = 0.00666 L V^{1.90}$ agrees well with the friction drop for this size of tube as reported by the experiments of Saph and Schoder * on friction losses in brass tubes.

GRAPHICAL DESIGN METHODS.

The design of a feed-water heater or other form of heat-transfer apparatus consists of determining the proper balance among the factors: tube length, number of tubes, velocity, total surface, friction drop, and temperature rise. The design methods explained herein are based upon the theoretical and experimental relations between these various quantities. The semi-logarithmic diagram of Fig. 14, showing relation between tube lengths, outlet temperatures and quantity of water per tube for a steam temperature of 240 degrees and an inlet-water temperature of 90 degrees, was constructed as follows:

The outlet temperatures for the various quantities per tube were determined from the curve of Fig. 12 and plotted opposite the unit tube length of 30.33 feet. Straight lines from these outlet temperatures and the inlet temperature of 90 de-

* Saph and Schoder, A. S. C. E., 1904, Vol. 5, page 253.

grees at zero tube length show the relation between tube length and temperatures at the stated velocities or quantities per tube. This diagram may be used to design for any desired

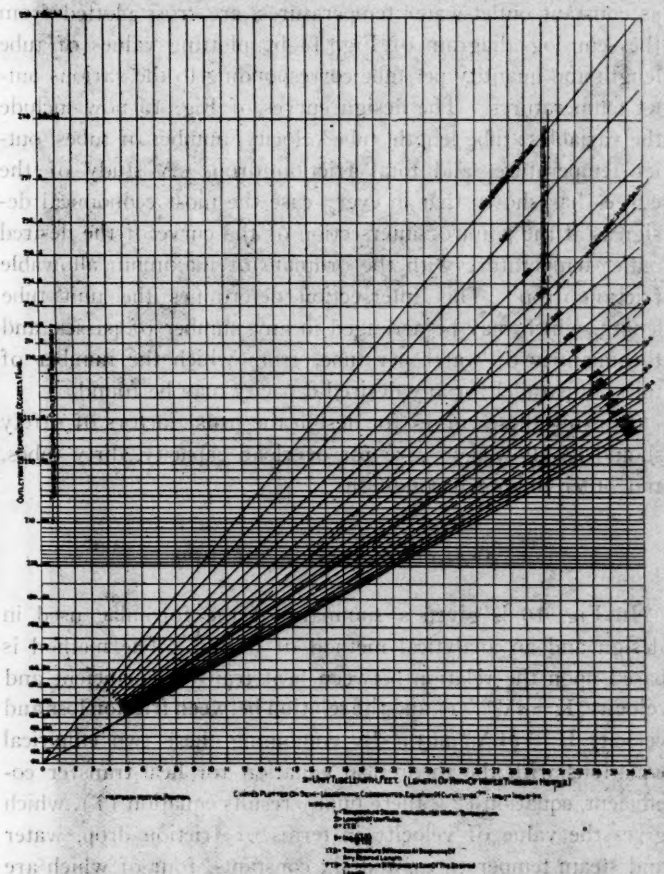


FIG. 14.—SEMI-LOGARITHMIC DESIGN FOR CURVES FOR HEATER WITH ⅝-INCH TUBES.

temperatures. In order to design for a correct friction drop also, the design curves of Fig. 15 are drawn. The straight lines of tube length *vs* friction drop at constant velocities are

drawn either from the formula for friction drop as determined from the test, or from the known friction drops for the unit tube length of 30.33 feet. The curved lines, marked as constant outlet-water temperatures, are cross plotted from the semi-log. diagram of Fig. 14 by plotting values of tube length and quantity per tube corresponding to the various outlet temperatures. The design curves of Fig. 15 now include the variables: tube length, tube velocity, number of tubes, outlet temperature, and total friction drop. A study of the curves has shown that in every case the most economical design is at the point of intersection of the curve of the desired outlet temperature with the ordinate of maximum allowable friction drop. This intersection determines the unit tube length, which may be arranged in any number of passes, and the quantity of water per tube, from which the number of tubes in parallel for any desired capacity may be found.

In using these curves for design the usual factors of safety should be allowed to care for overload capacity, dirty tubes, and other unknown conditions.

ANALYTICAL DESIGN METHOD.

In Fig. 16 is given a summary of the formulae used in design and an analytical method of design. The method is based upon the relation between heat transfer coefficient and velocity, $K = aV^n$...(1), and the relation between friction loss and velocity $H = bLV^m$...(3). By combining these two empirical equations with the theoretical equation for heat-transfer coefficient, equation (2), there finally results equation (7), which gives the value of velocity in terms of friction drop, water and steam temperatures, and six constants, four of which are experimental and two of which, c_1 and c_2, depend only upon the sectional dimensions of the tubes used. The values of the constants for ⅝-inch standard tubes, No. 16 B.W.G., as determined from several tests of heaters, are as follows:

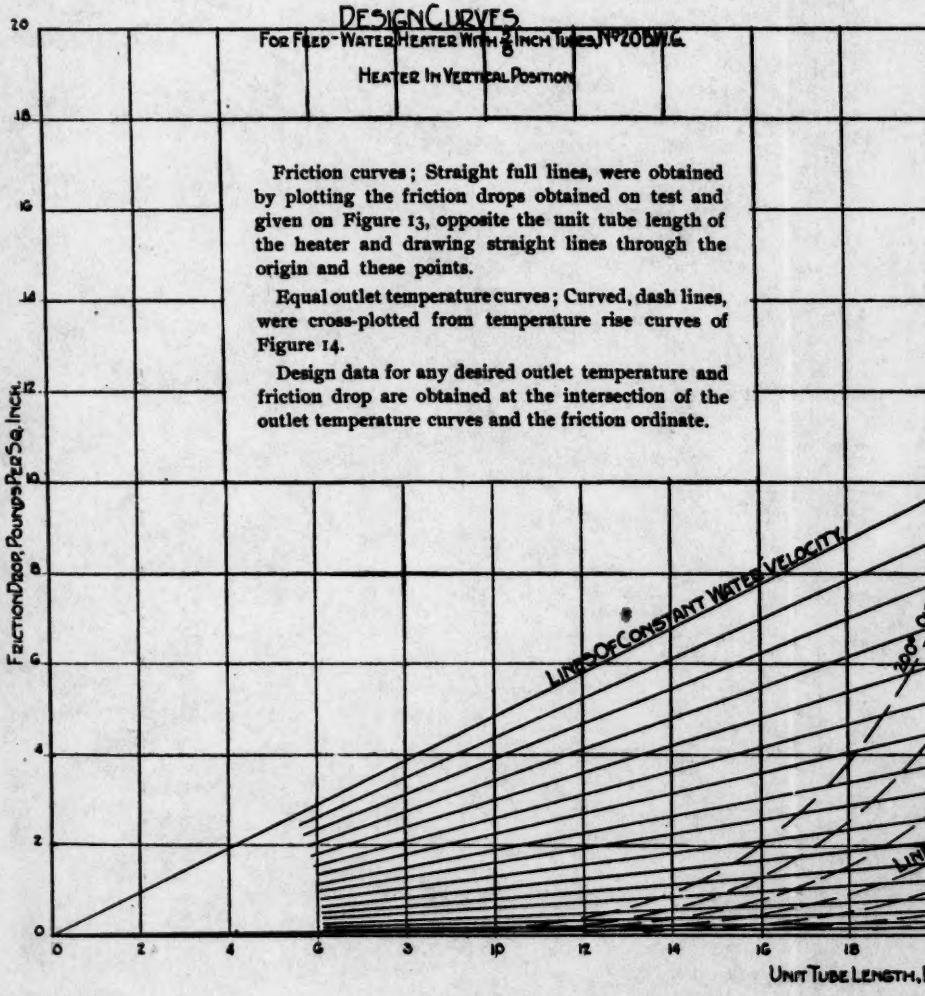
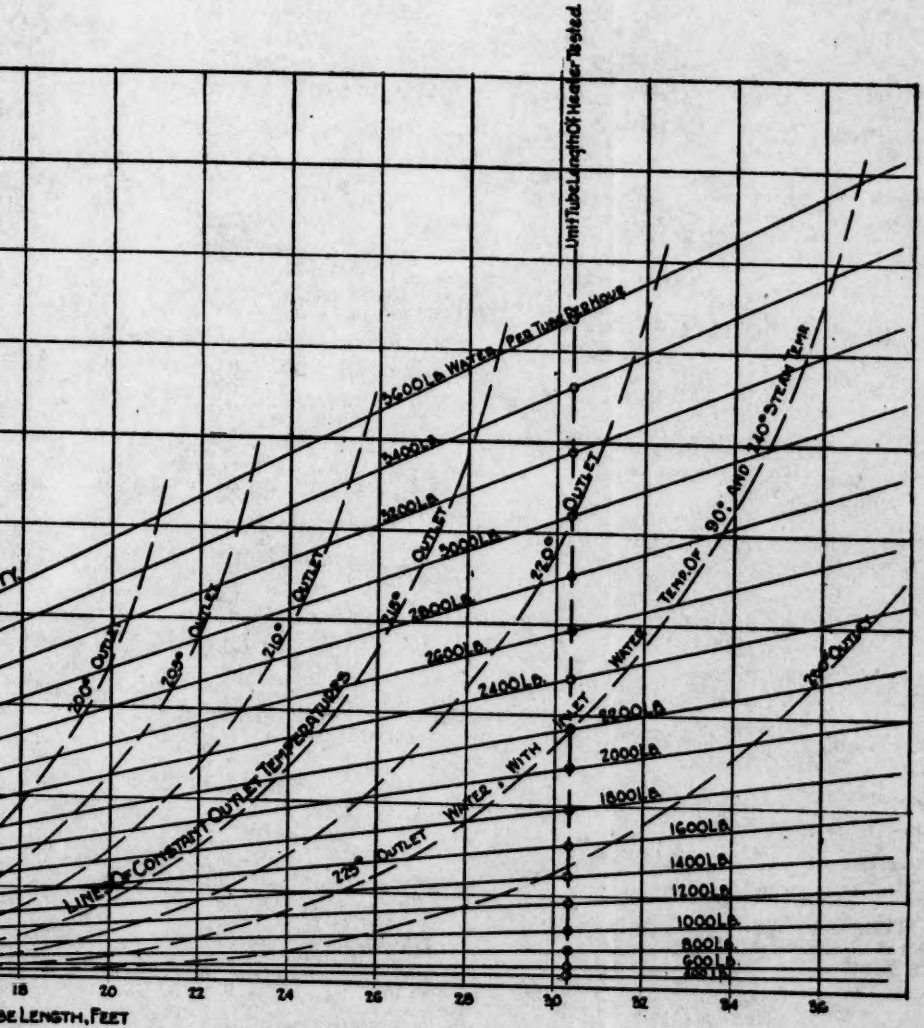

DESIGN CURVES
FOR FEED-WATER HEATER WITH 3/8 INCH TUBES, No. 20 B.W.G.
HEATER IN VERTICAL POSITION
Friction curves; Straight full lines, were obtained by plotting the friction drops obtained on test and given on Figure 13, opposite the unit tube length of the heater and drawing straight lines through the origin and these points.
Equal outlet temperature curves; Curved, dash lines, were cross-plotted from temperature rise curves of Figure 14.
Design data for any desired outlet temperature and friction drop are obtained at the intersection of the outlet temperature curves and the friction ordinate.
FRICTION DROP, POUNDS PER SQ. INCH.
LINES OF CONSTANT WATER VELOCITY
UNIT TUBE LENGTH,
0
2
4
6
8
10
12
14
16
18
20

FIG. 15.—DESIGN CURVES FOR HE



FOR HEATER WITH ⅝-INCH TUBES.

$n = 0.5$ to 0.6

$m = 1.70$ to 1.90

$a = 300$ to 340

$b = 0.007$ to 0.011

$c_1 = 305.8$

$c_2 = 0.1636$

ANALYTICAL METHOD OF DESIGN FOR FEED-WATER HEATERS.

K = Heat Transfer Coefficient, B.t.u. Transferred Per Hour, Per Sq. Ft. Heating Surface, Per Degree Mean Temperature Difference.
V = Velocity, Feet Per Second.
W = Pounds Of Water Per Hour Through Heater.
S = Total Heating Surface In Heater, Sq. Ft.
N = Number Of Unit Tubes In Heater.
L = Length Of Unit Tube, Feet.
H = Friction Drop, Pounds Per Sq. Inch.
t_s = Temperature Of Steam In Heater, Degrees, Fahr.
t_i = Inlet Water Temperature, Degrees, Fahr.
t_o = Outlet Water Temperature, Degrees, Fahr.
a, n = Heat Transfer Coeff. Constants Determined From Test Of Heater.
b, m = Friction Constants Determined From Test Of Heater Or Other Sources.
C_1 = Constant Depending On Inside Cross-Sectional Area Of Tube Used.
C_2 = Constant Depending On Outside Perimeter Of Tube Used.

$$K = aV^n, \quad (1) \qquad K = \frac{W}{S}\log_e\left(\frac{t_s - t_i}{t_s - t_o}\right), \quad (2)$$

$$H = bLV^m, \quad (3) \qquad L = \frac{H}{bV^m}, \quad (4)$$

$$W = C_1VN, \quad (4)$$

$$S = C_2LN, \quad (5).$$

$$C_1 = \frac{\pi \times d_i^2 \times 62.3 \times 3600}{4 \times 144}$$

$$C_2 = \frac{\pi \times d_o}{12}$$

In Which:
d_i Is Inside Dia. Of Tube Used, Inches.
d_o Is Outside Dia. Of Tube Used, Inches.

Substituting Values Of W, S, And K From Equations (4), (5) & (1) In Equation (2) We Obtain:

$$aV^n = \frac{C_1VN}{C_2LN}\log_e\left(\frac{t_s - t_i}{t_s - t_o}\right) = \frac{C_1V}{C_2L}\log_e\left(\frac{t_s - t_i}{t_s - t_o}\right) \text{ Or,}$$

$$V^{1-n} = \frac{aC_2L}{C_1\log_e\left(\frac{t_s - t_i}{t_s - t_o}\right)}, \quad (6)$$

Substituting Value Of L From Equation (3) In Equation (6) We Obtain:

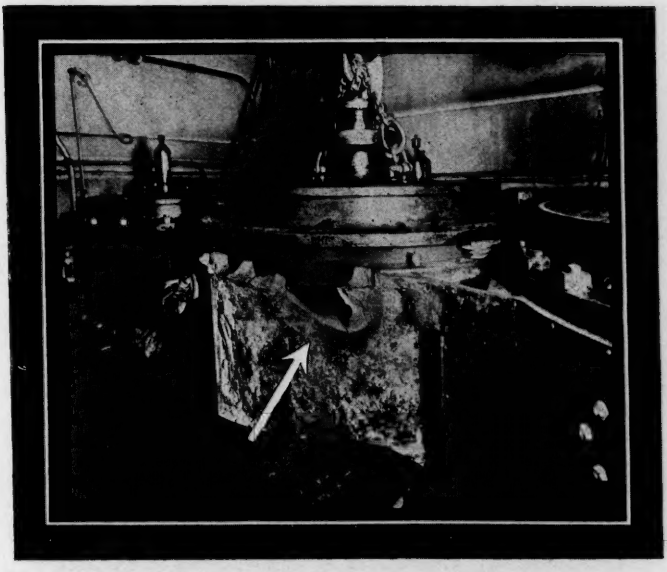
$$V^{1-n} = \frac{aC_2H}{bC_1V^m\log_e\left(\frac{t_s - t_i}{t_s - t_o}\right)} \text{ Or,}$$

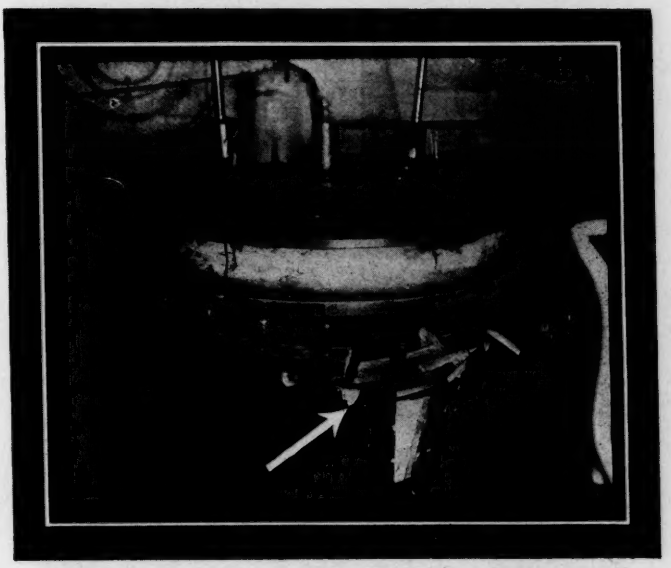
$$V^{1+m-n} = \frac{aC_2H}{bC_1\log_e\left(\frac{t_s - t_i}{t_s - t_o}\right)}, \quad (7)$$

Values Of L, N, and S May Now Be Determined By Substitution From Equations (3), (4) And (5) Respectively.

FIG. 16.

The formulae given in Fig. 16 may be used to design feed-water heaters or any similar apparatus with any type or size of tube for which the values of the six constants enumerated above are known. The proper velocity is determined from equation (7), and the value of tube length and number of tubes then determined from equations (3) and (4), respectively.





DAMAGED MACHINERY OF "BULGARIA," "RHEIN" AND "NECKAR."

—Courtesy of "Power."

DAMAGED MACHINERY OF "BULGARIA," "RHEIN" AND "NECKAR."

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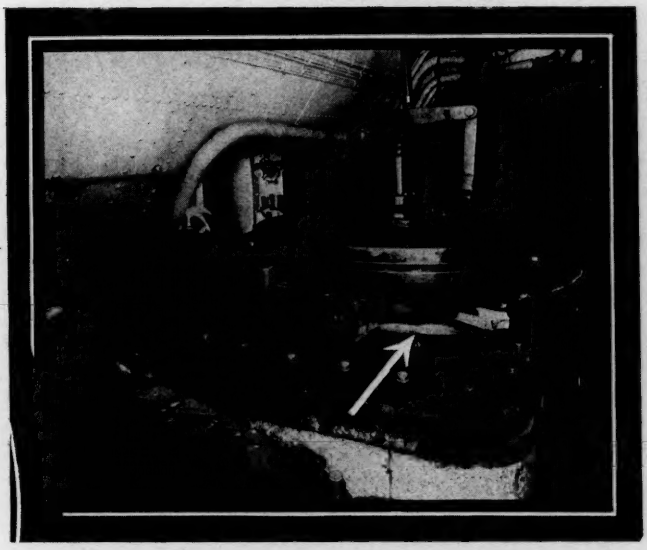

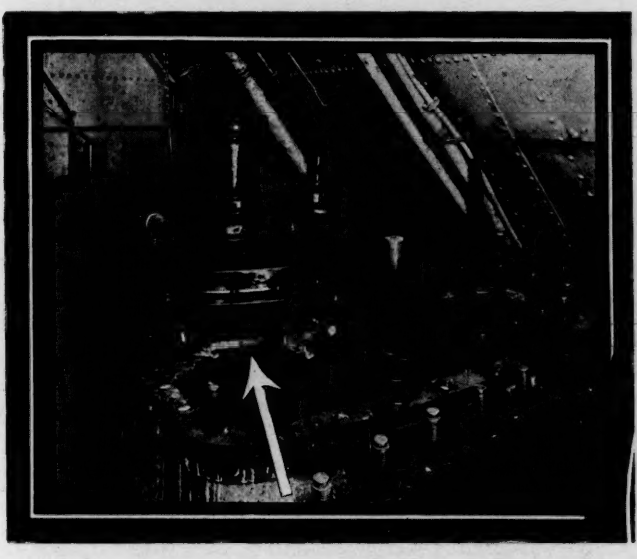

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DAMAGED MACHINERY OF "BULGARIA," "RHEIN" AND "NECKAR." —Courtesy of "Power."

VANDALISM.

Through the courtesy of "Power" this Journal is able to reproduce the first published photographs of the damage wrought on the machinery of the German vessels *Bulgaria*, *Rhein* and *Neckar*, recently taken over by our Government. The eight pictures on the preceding pages give some idea of the manner in which the cylinders and valve chests were wrecked.

Cylinders and valve-chest covers were removed and the cylinders and chests were wrecked by hammers and rams. In the boiler rooms thermit was used in at least one vessel. The principal boiler damage was caused by lighting fires in dry boilers. A casual survey might lead one to believe that destruction was thorough and beyond repair. If the directing mind believed the measures adopted would permanently disable these vessels, it failed to allow credit to American engineering ability and the patriotic attitude of our industrial establishments. The large railroads have taxed the capacity of their machines and repair shops to furnish equipment and skilled men to take care of some of this reconstruction.

The actual work has been done by electric and oxy-acetylene welding and by patching. While the use of welding in the repair of cast iron is common in railroad practice, it is not well known in the marine world, and its use is considered quite a departure in marine practice and, no doubt, will be followed with considerable interest.

Former practice would indicate that renewal of cylinders was necessary, with the attendant large loss of time and great expenditure of public money. On the contrary, with few exceptions, practically all cylinders have been reclaimed and will soon be in service.

THE JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS hopes soon to be able to publish a series of articles dealing with the methods of repair on all ships taken over by the Government as conducted by the several engineering officers in charge of the work.—THE EDITOR.

NOTES.

NOTES ON THE CONSTRUCTION OF TURBINE PUMPS.*

By Alan E. L. Chorlton, of Lincoln, Member.

The turbine has become so highly successful as a pumping engine for duties of all description that no apology is needed from the author for again returning to the subject and offering a few practical notes on the design and construction of its detailed parts.

It will be convenient to consider the subject under the headings of the various component parts of a pump. These are:

I. *The Stator*, which consists of (I. *a*) the casing or housing, and (I. *b*) the guide vanes or appliance for converting velocity energy into pressure energy.

II. *The Impeller*—considered separately.

III. *The Balancing Appliance*—hydraulic or mechanical.

IV. *The Rotor*, considered as a whole, and including the spindle with its projecting sleeves, impellers, and in most cases the balancing appliance.

V. *The Bed*, and other details, bearings, stuffing boxes.

Of these components III., IV. and V., broadly speaking, are the factors governing reliability and resistance to wear, and I. *b*, and II. are those determining the efficiency.

The author proposes to deal at greater length with III. and IV., for the reason that they have probably been less discussed than the others, also their practical importance is of the greatest moment, and he cannot deal with the whole subject adequately within the confines of a paper written during the present time.

The chief points only of items I., II. and V. will be briefly reviewed:

I.—CASING.

The individual impellers of a turbine pump revolve in chambers or cells containing the outward-flow guide passages, and the return conduits for the water; these chambers may be part of a whole in which the outer body is cast in one piece, or an aggregation of a number of distinct cells without any outside envelope. There are therefore two main types:—(1) The Osborne Reynolds or divided type, and sometimes called the "ring type." (2) The Sulzer, integral or one-piece type, sometimes called the "cylindrical" type. Recent American practice provides a variation of this type with the housing in halves divided on the horizontal center line. The two main types are diagrammatically illustrated by Fig. 1 (*a* and *b*) and Fig. 2 (*a* and *b*) with variations in Figs. 3 and 4.

The Osborne Reynolds pump and its evolution was dealt with by the author in collaboration with Dr. E. Hopkinson, in some detail in a paper read before the Institution in January, 1912. This type was followed at a later date by the integral one of Sulzer, which had, for a period, a considerable vogue. An examination of the present-day practice of various makers will prove, however, that the ring type has ultimately

* The Institution of Mechanical Engineers.

proved the preferred one. The Reynolds pump has always used a separate cell for each impeller, and it is therefore correct to say that the ring type of pump owes its inception to this country. The continental form of the series turbine pump is due to Sulzer—first pump 1896—and in this type a monoblock housing was used for all the chambers or cells, the guide vanes being inserted from the end.

Before dealing with either type or variations arising out of them it is advantageous to consider the essential functions of the chamber or housing of a turbine pump. Primarily, each cell consists of (1) the outward-flow guide passages in which the kinetic energy of discharge from the turbine impeller is converted into static head, and (2) the return-water passages back to the center for conduction to the next impeller; a complete housing is a collection of such cells. Obviously, in the design, commercial considerations must have a material guidance on theoretical claims. Against the requirements for best theoretical conversion of kinetic energy must be matched the allowable limits of dimensions conformable with commercial possibilities, and, in the interests of efficiency, special attention must be paid to the arrangement and dimensions of the divergent channels or guide passages. As is well known, the form for guide passages is represented diagrammatically by Fig. 5, and this passage must be disposed in some form to lie conveniently in the desired casing. The general character will be either a simple outward-flow type in one plane, Fig. 6, or a mixed type outward and axial in two planes—see Fig. 7 (*a* and *b*) and Fig. 8.

The divergent angle of guide vane, Fig. 5, for best efficiency was shown by Professor Gibson to be 10 degrees to 11 degrees. In many cases such a small divergent angle leads to a large overall diameter guide vane in order to give a sufficiently reduced speed of water to permit reversing its radial direction, and the result is a very heavy casing in consequence. For this reason divergent angles of 15 degrees are commonly found in practice. Evidently a more efficient pump will sometimes be heavier and more expensive than one less efficient, and commercial considerations must provide the final deciding factor between efficiency and weight. There is, of course, a school of design which believes in dealing with a proportion of the velocity conversion in the wheel itself, thus leaving less to be dealt with in the guide passage; the extent, however, to which this method can be used for weight saving is very small, if any.

The various assemblies of passages may clearly be grouped into:—(A) Tangential and radial with return radial—see Fig. 6 and Fig. 1 *b;* (B) tangential and spiral—see Fig. 7 (*a* and *b*) and Fig. 8; (C) combinations—see Figs. 9, 10 and 11. The Osborne Reynolds pump—1887 and 1875 type—employed an early form of the (B) assembly, and has adhered to this type up to the present day, various improvements being embodied from time to time, in some of which the author was concerned. On the Continent Messrs. Sulzer introduced in 1896 design (A), and the author believes they have made little departure from the type beyond a considerable simplification of their early arrangement of passages, Fig. 2 *a*.

Speaking generally, combination designs (C) are not so efficient as the simpler types (A) and (B), owing probably to the hydraulic loss through changing the radial direction of the water at high speed. From a works construction point of view, "ring casings," Fig. 1 (*a* and *b*), are the most economical, and in practice give high efficiency. In the form similar to that shown in Fig. 12, the author some years ago was able to mold and cast ring casings without cores, machine molding being adopted, and the cost per chamber coming out at a very low rate.

For "cylindrical" casings, Fig. 2 (*a* and *b*), a complete pattern is

required for each size and variation in number of chambers. Its accessibility, however, and ease of dismantling is sometimes considered to be greater than with the "ring" type, though this is to a certain extent a matter of opinion. A great drawback to casings containing separate cells is that, on account of the sliding fit between the intermediate pieces and casing, an unknown amount of leakage constantly takes place between the cells. With a "ring" type of pump, leakage is instantly detected and can be remedied. As a commercial proposition the author unhesitatingly favors the divided or "ring" type of casing, and when properly carried out has met no difficulty with it in practice.

Before leaving this part of the subject, a word about the finish of guide-passage surfaces is necessary. For best efficiency the throat of the passage at least, if not the entire passage, should be gun metal or bronze, as iron does not preserve a sufficiently good surface for high velocity conditions. Common practice is to provide only three sides of the guide passage in bronze, Fig. 13, but this can only be defended on grounds of cheaper first cost. For best results a bronze plate should be provided to box in the passage, the plate being attached to the guide-vane casting or dowelled to the casing. Guide vanes are sometimes cast completely boxed in, and this method necessitates hand finishing of the passage by file and scraper; open vanes, however, lend themselves better to cleaning out and accurately finishing either by hand or by machining. A good smooth surface is essential for the best results, and will always justify the increased cost.

The method adopted of securing guide vanes from rotation and vibration must be a thoroughly sound one or trouble will result.

II.—IMPELLERS.

Impellers are either:—(1) Single entrant or (2) double entrant. The first is almost universally in use for multicellular pumps, and the second almost exclusively for single-chamber pumps; it is only proposed to deal with the single-entrant form. Multicellular pumps use the single eye wheel in three forms:—(*a*) Unbalanced, unequal side area, one rubbing shoulder, Fig. 14; (*b*) unbalanced, equal side areas, two rubbing shoulders, Fig. 15; (*c*) balanced—on paper, Figs. 16 and 17. There are other types, mixed flow, radial and axial, &c., but these are not used to a sufficient extent to be worth including in this paper.

Type (*a*) may be said to be one now generally used, that is, the preferable design; type (*b*) has certain advantages for machining, &c., but probably requires a stiffer shaft. Also, having no central supports, it tends to rotate the suction water of the next impeller and may be attended by greater stage leakage due to the increased annular running clearance; type (*c*) has the *paper* advantage of being in balance; actually it is a poor approximation to a balance. Disturbing factors are set up by differences in side pressure on the impeller due to differences in volume and surface form of the water contained on the two sides of the impeller, Fig. 18; differences of quantitative leakage through the two shoulders; and high-pressure leakage *into* one side of the impeller from the stage above, and leakage *from* the other side of the impeller to next low-pressure stage below. Therefore, in practice, it is necessary to provide an additional end-balancing device of the hydraulic type, or a mechanically-positioning fitting such as a thrust collar or ball-bearing; a method, especially for mine usage, not to be recommended.

The internal design of all impellers is governed by the same controlling features:—(1) The entrance or inlet angle of vane; (2) the delivery or exit angle of vane.

The entire design must, while based on these considerations, consult

the convenience of the workshop to the utmost degree possible without departing from required dimensional accuracy. The standing difficulty with turbine pumps, from the manufacturing point of view, is their constant variation to meet the infinite number of conditions of varying head, speed and quantity encountered in practice. Whatever efforts are made, it seems impossible to keep to a small number of standard impellers if the highest efficiency is to be reasonably well reached each time. Efficiency, it should be noted, is really the prime factor in design and not apparent first cost, for the pump is very often driven by an electric motor of greater value. If, therefore, by the use of a pump of higher efficiency a reduction is effected in the necessary power and size of motor, the combination will generally come out cheaper.

The maker who elects to change his impellers and diffusers to suit the demands of the inquiries as they come in, will obtain a higher percentage of orders than one who has standardized his, even though the latter may have reduced costs by making in large quantity. In considering if in any way it is possible to meet the designer's requirements without losing all the advantages of repetition manufacture, we may first take the inlet angle, This depends on the resultant of the—supposed—radial flow of the entering water and the peripheral speed of the inlet tip itself, and it usually varies between 15 degrees and 30 degrees. If we select to use always 15 degrees we might suffer in some cases to the extent of 3 per cent. or 4 per cent. If we take two sizes, 15 degrees and 25 degrees, and make a liberal provision of inlet width, or, as some designers phrase it, make an adequate allowance for "weir coefficient" at entrance, we shall only drop perhaps 1 per cent. in exceptional cases; so it seems possible to do something in the way of standardizing the inlet angle.

One might refer, in passing, to some of the current ideas concerning the inlet angle and the condition of the water at entry to the impeller. Some designers have maintained that very great accuracy of the inlet angle is vital to high efficiency, in spite of the fact that in the ordinary unobstructed eye it is impossible to say what the absolute velocity and direction of the incoming water is, and, therefore, to estimate precisely what the correct inlet angle should be is impossible. Further, the behavior of the incoming water varies with every rate of flow, and the only way to foretell its condition would be to insert inlet guide vanes; this, however, in such experiments as the author is aware of has proved an objectionable practice and only introduces further losses in the pump. Another school holds that it is an advantage purposely to introduce a forward whirl in the water at the eye with the object of helping the water into the impeller. If such an initial whirl is used, the work necessary to create it is necessarily done by the pump itself, and it is, therefore, questionable if this does not entirely overweigh any possible gain.

This discharge angle depends on the resultant of the peripheral speed of the wheel and the radial outflow, the variation of which is so great it seems impossible to devise any standardization. To meet quantity variation two widths of impeller may be used, and to meet required speeds of revolution it is usual to allow a small percentage variation in the impeller diameter to suit special demands; this latter, however, is conveniently done without pattern or casting alterations.

The foregoing considerations lead to the conclusion that nothing less than a special core box will be required for each case, and it is quite usual to meet the difficulty in this way. Evidently, however, a most valuable appliance would be a special form of core box, which in itself was more of a standard and could be adopted for manufacture in quantity; this result might be effected by fitting in vanes of a flexible nature, so that the necessary alteration and adjustment would not be great, and the whole outfit would come out cheaper than if a patternmaker had to

build a fresh box and fittings for each new demand. As the accuracy and smoothness of the impeller is of great importance, the author has always looked to machine molding, and to a special machine like a wheel-molding machine, for the purpose; he has not, however, as yet, arrived at the final design for such.

The surfaces of the impeller, both inside and out, play an important part in the efficiency of a turbine pump. The smoother the outside surface the less the power lost in disc friction, and the less the power wasted in revolving idle "dead water." An interesting point to note is that the greater the speed of revolution of the "dead water"—that is, the greater the power absorbed in this way—the less the leakage from the periphery of the impeller; but a little consideration will show that greater overall economy is gained by reducing the wasted power in idle revolution to a minimum. As regards the internal smoothness of the passages of the impeller, it is usual practice to clean up the surfaces as well as possible with file and scraper. Impellers have been built up with one loose side so as to permit of machining or more effectively cleaning the interior; and it is evident that with individual impellers producing high heads the results would well repay the extra cost. Objections to this practice are the difficulty of making attachment of the two parts and the extra weight necessary to provide attachments.

III.—BALANCING APPLIANCE.

The experienced designer knows, from practical knowledge, that even if a group of impellers, or even a single impeller, is theoretically balanced by equivalent areas subject to pressure, it will not in actual practice be free from all end thrust. Apart from variable leakage at the two sides of an impeller, which, by the way, is the principal cause of end thrust, we often have variable side surface both of the impeller and the cell chamber. If the conditions of capacity or surface of the two clearance chambers vary, the resulting pressures will vary and an axial thrust is set up. Professor Gibson has investigated the effects on efficiency—power lost—due to varying the side clearance of the impeller, and he took readings of circumferential pressures set up by revolving discs, but he apparently did not plot out the resultant effect in end pressure. This interesting problem is discussed in a paper by F. Zur. Nedden, where use is made of the results from both Professor Gibson's and Professor Unwin's revolving-disc experiments. The general effect of the gyratory pressures set up is indicated in Fig. 18. It is seen that the speed of the revolving "dead" water at the two impeller sides sets up pressure in opposition to the leakage pressure from the impeller tip, as already pointed out; the faster the water revolves the greater the resistance to leakage, and the less the resultant pressure due to that leakage. It can readily be seen, therefore, that the effect of the chamber at one side of a wheel having greater capacity than the other, or having more obstructions in the way of ribs, pockets or exceptional roughness, &c., plays some part in the determination of end thrust.

All the earlier turbine pumps employed some form of thrust bearing, either of the collar or the ball type, to keep the impellers in correct alignment with the guide passages, and it was not until much trouble had been experienced with these that hydraulic control was ultimately adopted. Experience showed that much higher end pressures were set up than were ever anticipated, but for a period mechanical devices were persisted in, improvements being made attempting to withstand the excessive loads, and scant notice, it would appear, being taken of water-turbine practice where for many years it had been the custom to relieve axial thrust hydraulically. Fig. 19 illustrates a hydraulic balancing device in use on Francis turbines, and which, if properly proportioned, is automatic, and

thus embodies all the essential points of many present turbine-pump balancers. One should bear in mind that the relation between the calculable axial thrust and the thrust realized in a water turbine is much closer than in a centrifugal pump of the multi-stage type, the reason of this being that the disturbing factor in a high-lift pump is the leakage from stage to stage.

The well-known application of turbine pumps to high lifts by Messrs. Sulzer, at Horcajo Mines in 1898, was carried out with back-to-back impellers, and a ball thrust bearing was provided to take the end thrust—in one direction only—which is inseparable from this arrangement of impellers. In 1901 Professor Rateau was manufacturing pumps with end thrust approximately eliminated hydraulically by his well-known method of shroud reduction, and provided with a balancing piston, Fig. 20. This method was not automatic, and the pressure on the piston could only be adjusted by means of a hand-operated throttle valve. However, this was the first step, and the automatic control of the necessary pressure on the balancing device, and determined by the end movement of the spindle, followed as a matter of course. The development of the differential type of balancer carried out under the author's direction in 1913, extended over several years, progressing step by step from a hand-adjusted needle valve, Fig. 21, regulated to produce the required balancing pressure, to a mechanically-operated needle valve, Fig. 22, actuated by the axial movement of the spindle, and then through several forms of rotating throttle valve disposed on the spindle itself and actuated by the same means. Further reference to the final balancing appliance—diagrammatically shown in Fig. 23—will be made shortly.

In 1906 Messrs. Sulzer brought out their device for relieving thrust shown in Fig. 24, and the principle used had been followed by many other makers and introduced with many variations. The action of the Sulzer device is simple; it comprises a single plate with a throttling device, through which the pressure water has access to the plate. The area of the plate is sufficient when acted on by a somewhat throttled delivery pressure to overcome the tendency of the rotor to travel in the opposite direction. As the pressure gradually rises before the plate, its increased power carries the rotor to the right until further motion is arrested by the escape of the increased pressure through a widening annular space between the plate and its facing seat. An ultimate running position is reached when the leakage through the clearance between valve and seat is sufficient to maintain a pressure at the back of the plate, equal to the set of the rotor in the opposite direction. It will be seen that the balance is quite automatic in its control, end movement in either direction taking place until stability is reached.

Without going further into the detailed evolution of automatic control of the hydraulic end balance of the turbine, it is sufficient to say that the devices adopted resolve themselves into two basic forms:—(*a*) Single acting—simple—of which the Sulzer, Fig. 24, is the general type; (*b*) double acting—differential—of which Fig. 23 is the type, and Fig. 13 a modification.

Other single-plate forms are shown in Fig. 25 and 26. In the first of these the throttling agent, a small nipple, is removed to the discharge side of the plate, thus giving, in a measure, a double action to the plate; the quick action and limited travel due to the second face of the differential type is not obtained, however, and there is a possible disadvantage in the small aperture when dealing with impure water. The second example combines some of the points of both forms of balancer, but as the pressure in the spent-water chamber is always augmented by high-pressure leakage into the low-pressure side of the plate, the device is, obviously, relatively extravagant in leakage water, and must therefore be considered inefficient.

The general effect of wear on single-plate balancers is increased leakage, the outcome of which is further end movement in one direction until finally the impeller discharges do not match their guide-vane entrances but become off-set. This tendency in hydraulic balancers led the author to devise form "*b*," or the double-action control with which such difficulties are overcome.

Fig. 23—also in Fig. 12—finally shows the present standard double-plate or compound differential balance arrangement. The balance piston and regulating valve are combined, and form a balance disc with two active throttling surfaces floating between two fixed seatings with a minimum clearance. All the working faces are arranged vertically so as to be independent of possible vibrations of the rotary system. The pressure water, escaping from the rim of the last impeller, passes through the holes *h* into the pressure chamber *m*, from which it escapes along the throttling surface of the balance valve into the regulating pressure chamber *k*, and from there, past the small throttling surface into the escape pipe *p*, leading preferably to an open drain, or back into the suction chamber of the pump. The double-plate balance appears to be the most sensitive automatic balancing device in practical use, for by it, closing up of the plate on the large throttling face causes an opening out on the small one, thus multiplying the balancing effect; a movement of a thousandth part or so of an inch being all that is required to counterbalance the most extreme and sudden changes of the hydraulic equilibrium of the pump. The other and most important feature of this auto-plate balance is that it always maintains the central position of the impellers, even if the throttling surfaces become worn, as is invariably the case when pumps are working on gritty water.

A continental form arising out of this type is shown in Fig. 27, and it is obvious that the same remarks concerning extravagance in leakage water apply to this example as to Fig. 26. The arrangement of the balancer at the end of the spindle outside the external bearings is not a good one, for the reason that a high-pressure gland at the delivery end of the pump is still required, and an additional gland also for the balancer housing.

A single-plate balance made by the same firm as the above is shown in Fig. 28, and this, too, is open to the same objection as regards disposition on the spindle.

It is usual to make hydraulic balancers with renewable faces at the points of close running, so that the appliance can be readily repaired after wear has taken place. As regards the best material to be used for the renewable part, some considerable experience is needed before a decision can be made. Gun metal was probably the first material to be used, but cast iron, cast steel, and hard bronze have all been tried, the present practice usually being to fit hard bronze. The ideal material is one which is hard and "short," and has a very low coefficient of friction when working in water; a material which "drags" is quite unsuitable, and apparently an incorrodible iron, one containing a high percentage of silicon or a nickel steel, seems to best fit the requirements. Non-metallic substances, such as red fiber, woodite or dexine, have been considered, but any material which becomes slightly absorbent after long immersion is useless.

Sometimes difficulty is experienced in starting up large turbine pumps on account of the large diameter balance rings being held together in metallic contact instead of being apart in their usual running position; the result is an abnormally high starting torque for the motor and probable damage to the faces of the balancing device. When a turbine pump with differential balancer is "stopped," the rotor will always take up a position with the balancing faces apart if the waste-water outlet, Fig. 29, is taken to an open drain; however, if the spent water is led

back to the suction pipe of the pump and a foot valve is provided at the end of the suction pipe, the whole rotor is forced towards the suction end of the pump as soon as the foot valve closes and the head pressure comes on. The reason for this is that the area of the balancer on the outlet side is greater than on the inlet side by an amount equal to the diameter of the spindle For this reason the waste water from a balancer should not be connected to the suction pipe of a pump, or means should be taken in the pump to equalize the areas subject to the static hydraulic pressure, Fig. 12.

IV.—THE ROTOR.

As is well known, the internal design of a multi-stage turbine pump involves a rotor comprising a number of impellers keyed on to a shaft, the shoulders or bosses of which abut on each other and are secured and maintained together by double nuts or the like at, or near, the ends of the shaft. Such portions of the shaft as are not covered by the impellers, but are subject to the action of the pumped liquid, are protected by sleeves which fit up to the impeller bosses and thus make a complete sheath outside the shaft itself, Fig. 1 (*b*) and Fig. 2 (*a* and *b*), &c.

This rotor revolves within the guide chamber and housing, each impeller running at some points in the closest proximity to the partitions dividing the pressure stages and return guides.

These so-called running joints or neck rings must be so maintained by design and construction as to secure in continued service the smallest possible hydraulic leakage back from stage to stage, or mechanical loss by rubbing friction, both of which affect the efficiency and wear and tear of the pump. As a question of practical importance the design of this combination probably follows next to that of the successful automatic hydraulic balancing of the axial end thrust. We have in it the question of the deflection of the pump shaft; the reinforcing effect of the surrounding impellers; the support afforded by the bushes, if any, between each stage, and the loss by bush friction—wear; the form of neck ring to reduce leakage; and the effect of keyways and keys, &c., all independent of each other and affecting the whole.

The ideal condition is that of a rotor supported in lubricated bearings with a shaft of such sufficient stiffness between supports that the deflection, under all possible running conditions, is less than the clearance allowed at the neck rings and intermediate bushes, so that no contact takes place between the rotating and the fixed members, this clearance at the neck rings and intermediate bushes being kept down to the smallest possible limits. It will be seen that it is impossible to present mathematically the exact conditions with such a number of incalculable factors to take into account. For instance, it may be thought that a not inconsiderable reinforcing effect is obtained by the combination of the impellers outside the shaft as a whole. In practice, however, it is not advisable to lock up the impellers together shoulder to shoulder against each other by the end nuts in order to get the advantage of the reinforcing effect; because, due to, amongst other things, the slight inaccuracies of manufacture in squaring the shoulders of the impellers, sleeves, &c., the result would be to throw the combination out of truth, that is to distort the shaft. It might be contended that this need not be so, but even then provision must be made for dismantling in a mine or similar place with its obvious attendant disadvantages, and the possibility of rough treatment taking place; furthermore, as a safeguard against heating up and consequent expansion of the outside combination—bronze—against the internal—steel—shaft due to accidental contact with the inter-

mediate bushes, running, perhaps, when the pump is empty, suitable expansion should be allowed for, and the impellers, to ensure this, must not be locked tight against each other.

The supporting effect of the bushes on the shaft, in passing through the diaphragm intermediate between the impellers, is very difficult to exactly allow for; it may be that the bushes will decrease the deflection of the shaft by a material amount, but it must not be assumed for one moment that they are bearings of such a nature as lubricated bearings. Intermediate bushes can only, in some cases, be considered as water lubricated supports which will act as such so long as a certain low surface pressure on them is not exceeded; if too great pressure comes on, heating takes place on account of the high speed of rotation. The effect of wear has also to be taken into account, for the hydraulic pressures at the two ends of these bushes are different, and there is, therefore, always a flow taking place through them, carrying with it any sedimentary or foreign cutting matter which may be in the water. The state of things causes the supports to wear more or less rapidly so as to become almost useless, the pump eventually arriving at a condition worse than if a suitable clearance had been allowed at the first.

The author believes that too much use is made of these intermediate supports in turbine pumping design. It is found in practice that an internal bearing to be successful must have the same water pressure at both ends, and must be properly lubricated with good grease; it then gives excellent results. We are thus left with the fact that we are practically dependent on the shaft itself for the necessary strength and stiffness to allow of fine internal clearance, and the importance of a good design which will economically give the minimum deflection of spindle at once becomes apparent.

The factors affecting the deflection of a turbine pump spindle are:—Static: (*a*) The weight of spindle and distribution of diameter change; (*b*) the weight and distribution of impellers, balancer, and parts; (*c*) the number and span of supporting bearings. Dynamic: (*d*) In the *dynamic* condition, other incalculable forces entering into the account, such as centrifugal forces due to the out-of-balance masses, and finally certain hydraulic disturbances.

(*a*) Should be as light as possible consistent with the necessary stiffness; in common practice spindles are practically parallel the whole length. (*b*) Impellers and balancer should be grouped together as closely as possible, only allowing sufficient space for the water passages between the stages; also, the entire weight should be brought as close to the supporting bearings as possible. (*c*) The span of supporting bearings plays such an extremely important part in the durability of the pump; in the possibility and preservation of fine clearances; in the whirling of the shaft and in the determination of the most economical size of spindle, that special notice will be taken of it. (*d*) A loaded shaft supported horizontally between two bearings will "sag," and when rotated must suffer bending at every revolution. Also, there are bound to be certain out-of-balance masses in the rotor due to keys, heterogeneous composition of material, and the unavoidable variation in thickness of castings, &c. In addition, a shaft has vibrational periods due to its length and diameter, the whole question of vibration being intensely complicated by the loading and supports.

In addition to the disturbing factors mentioned above, vibrations are set up from the reaction of the impeller vane passing the guide vane. It is usual to reduce the intensity of this disturbance by arranging an odd number of vanes in the impeller relative to the guide vane, and by setting successive impellers on the shaft in such a way that the point of passing vanes is progressive throughout the series, *i.e.*, a "lead"

is given. The resultant of these arrangements is to produce a vibrating influence, which may be torsional or combined torsional and transverse, of very high frequency and enfeebled intensity and which, when properly carried out, is free from practical disadvantage. The damping effect of the water film in neck bushes is appreciable in overcoming slight vibrations, as is conclusively shown by the different behavior of a spindle when running a pump dry and when running it filled with water. In order for a shaft to oscillate it must force liquid out of one side of a neck ring, and the retardation offered by this action has a noticeable effect. For this reason, quite apart from the lubricating effect gained, it is always a wise precaution to fill a pump with water before running it, otherwise if the shaft is not a stiff one seizing up will probably occur. The remedy for spindle vibration due to any cause whatever is always a stiff shaft.

The whole matter of whirling shafts has been examined by several investigators, prominent among whom is Professor Dunkerley, but it is not proposed to discuss the matter further than to refer to the general proportions affecting the critical speeding of whirling. We find that the length (l) of span of bearings, and the diameter (d) of the spindle have an effect varying as $d^4/^3$.

Turbine pumps do not usually run at speeds approaching the critical speed, but the maximum safe speed for any rotor will bear a direct relation to the critical speed, in the sense that it will be an equal factor for equal safety. The above relation shows then, that for a fixed critical speed a reduction in the span of the bearings results in a material reduction in the necessary shaft diameter. The effect of axial thrust—always present in a turbine pump—in lowering the critical speed of a spindle should always be borne in mind.

With a view to considering the effect of the spindle diameter and span of bearings on the degree of fineness of the running clearances, we will consider the extent of the deflections.

The following arrangements of bearings are possible and are illustrated in Fig. 30:—(a) Two outside bearings; (b) one outside and one inside bearing; (c) one outside and two inside bearings.

A typical example was taken, and it was found, after drawing out the different arrangements, that the ratios of the spans of the bearings supporting the loaded shaft were, for the three cases, 1.86, 1.44, and 1 respectively, calling the shortest span unity.

As the deflection of a circular spindle is proportional to l^3/d^4, the deflections themselves for the above spans are proportional to 6.5, 3, and 1 respectively, or, the necessary sizes of spindles to maintain the same deflections are proportional to 1.6, 1.3 and 1. These results may be conveniently tabulated:—

Case.	Ratio of span.	Ratio of deflections.	Ratio of spindle diameter to preserve constant deflection.
(1) Two outside bearings. Fig. 30a	1.86	6.5	1.6
(2) One outside and one inside bearing. Fig. 30b	1.44	3.	1.3
(3) Two inside bearings. Fig. 30c	1.	1.	1.

It will be noted that no allowance has been made for the buoyancy of the medium—water—in which the rotor revolves. The great importance of keeping the distance between bearings in a multi-stage turbine pump as short as possible is abundantly clear from the foregoing, for, with the more or less customary type, Case 1, Fig. 30*a*, with outside bearings at each end, it is seen we have a deflection of six and a half times that of a pump with two internal bearings. The direct result of this extra deflection is that the clearances between the outside of the eye of the impellers and the neck ring must be increased to allow of the impellers revolving without touching, and this increased clearance necessarily means a correspondingly increased leakage and loss of efficiency. Further, as has been shown, the passage of a large volume of water through the running clearance means increased wear, and a tendency to push the walls apart, and therefore force the impeller and shaft still farther out of the straight, with consequent friction against the adjacent neck ring surface.

Dynamic out-of-balance may—and very probably will—exist even though considerable care has been taken in the workshop to overcome it. This factor is a very important consideration, and calls for a very stiff shaft to resist its possible occurrence, because, if the shaft and impellers are thrown out due to this, then further clearance at the neck rings is necessary, or abrasion will result. The question of the reactive effect of an impeller discharging unequally round its circumference is capable of so many variations that the only practical way to overcome this also is a mechanical construction that will resist deflection—that is, a stiff shaft.

There is a further disturbing influence which is more active in smaller pumps, namely, the effect of the hand-packed and often ill-adjusted stuffing box and gland. This detail is always standard for one end of the pump, and sometimes found at both suction and delivery ends. Apart from the cutting effect of such a packing on the shaft or its sleeve, it also, by reason of its compression in screwing up, tends to set the shaft out of the true center, and so may intensify many other similar influences. The risk of outside bearings, separate or overhung, being slightly out of center, must also not be forgotten. For very many reasons, therefore, it is extremely important that the distance between bearings be kept as short as possible, and to do this successfully, internal, specially designed, large diameter, grease-lubricated bearings are essential.

Summarizing the effect of a long distance between bearings, we have:—(*a*) Increased deflection of rotor, (*b*) increased neck ring clearance, (*c*) increased leakage from delivery to suction, (*d*) increased wear thereby, and (*e*) decreased efficiency.

Turning now to the dimensions of the rotor, the table shows that the diameter of the shaft does not increase as fast as the length of span increases. However, a very serious increase in size—1.6 times—is entailed by the increase in span consequent upon changing from two internal bearings to two external bearings. Obviously increasing the diameter of the shaft will give the decreased deflection desired, but then it introduces a larger diameter of impeller boss and entrance eye, thus bringing about a corresponding drop in efficiency, as well as more leakage, owing to the larger leakage area consequent upon the increase in diameter of the boss and the neck rings. It is clear, therefore, that securing the stiffness of a shaft by increasing its diameter is an inefficient method, for, besides the increased area for leakage through the neck rings, we have much worse conditions of entrance in larger and more rapidly-rotating eye and inlet edge of the impeller vanes. The desirable condition, therefore, from all points of view is to use the smallest diameter of shaft with the shortest distance between the bearings.

Leakage between Rotor and Stator.—The leakage through the neck rings may now be considered under different conditions and clearances. The material of these rings is the first important consideration. It is obvious that a material should be adopted in the first place of such a type that it will not tend to drag or tear if accidentally touched or rubbed by the running impeller. Material of a hard, "short" nature, like cast iron, will meet this condition, and in practice the smallest clearance can be successfully run with such neck rings. They are, however, subject to corrosion, particularly with acid waters, as are frequently met with in mines.

The impeller, for the reason of susceptibility to corrosion, together with considerations of the necessary strength to resist the centrifugal effects of high speed, is usually made from phosphor bronze. We therefore frequently get in practice phosphor bronze impellers running in phosphor bronze neck rings, though it is obvious that dissimilar metals would be preferable. A phosphor bronze impeller with a cast-iron neck ring is a better combination, and will run with smaller clearances, but to get over the corrodibility of the cast iron it should be specially treated, or should be similar to that indicated for renewal balancer rings.

Soft metals will not run with fine clearances, for, if any rubbing is set up temporarily or otherwise, through dynamic out-of-balance, or want of alignment from any cause whatever, the metal immediately drags and the pump seizes up. The softer the metal the greater the clearance necessary. White metal neck rings have been tried, but were for this reason a failure. Lignum vitæ is possible, but owing to its uncertain expansion, a liberal clearance allowance must be made. It provides, however, a material to meet special conditions—mostly stay bearings in vertical shafts. Lignum vitæ is also frequently used in water turbines, tail-shaft bearings for ship's propellers, &c.

The form of the neck ring is in practice confined to four forms:—(*a*) Internal, (*b*) external, (*c*) vertical, (*d*) labyrinth—types of which are illustrated in Fig. 31. The first is the usual form in turbine pumps, the second is more rare, the third is practically restricted to single impeller low-lift centrifugals, while the fourth is a refined form after steam turbine practice, and is used in some high-lift pumps.

Between (*a*) and (*b*) there is little to choose, but for manufacturing reasons (*a*) is usually preferred. Clearances of a few thousandths of an inch per side can be run with a suitable shaft, and the leakage is very slight—less than 1 per cent. (*c*) The vertical form is used in pumps for low lifts, subject to dirty water and inferior attention. It allows of downward wear of the impeller without any rubbing surface coming in contact, which (*a*) and (*b*) would not. (*d*) is an attempt to reduce leakage further by a device of the kind used on the steam turbine.

For the reason that the labyrinth type deals with a non-expansible fluid like water, and not steam, it is far less effective than in the steam turbine, and, in fact, can only restrict leakage by the formation of eddies in the channels or grooves; shock losses at the sudden changes in area and direction, and the centrifugal effect of the rotating film of water in the narrow aperture. Its possible advantages in this way are more than negatived by its fragile nature, if it comes in contact with the impeller, such a contingency usually involving the eventual renewal of both parts. Types (*a*) and (*b*) only require a new neck ring, after wear. All types, however, are readily made with loose rubbing rings, which may be easily changed.

In practice it has been found that when grooves were cut on the impeller shoulders to secure a labyrinth effect, and at the same time the neck rings were left plain, that after running for some time the marks of the grooves were always found on the neck rings; the difference be-

tween the worn surface and the original surface often being as much as ten thousandths of an inch, showing in these cases a deflection of the shaft greater than the calculated value. In these particular cases the fact that the grooves appeared all round the ring to the same depth indicated that they were due to actual deflection, and not faulty alignment, the reasons for which may be found in the causes already enumerated and in mechanical inaccuracies. When interlocking grooves are used the difficulties of erection and dismantling are increased, unless the interlocking is in the plane at right angles to the axis of the pump. Any other arrangement is, of course, only possible with horizontally split pumps or with split or divided neck rings.

V.—THE BED AND OTHER DETAILS.

In the past less importance has been placed in the effect of the bed plate than is now the case, with the result that frequently very shallow depths were used. These early beds were much too weak to resist warping strains, due either to the unevenness of the foundation surface, the "pull" of holding-down bolts, or to the reaction of the torque from the motor end. It was a common practice to design bed plates in the form of two separate horns or branches, and in this case deeper sections were generally employed. This form of design, while undoubtedly stronger against bending, depends too much upon the skill of the plant erector and the solidity of the foundation to always give good results.

It must be obvious that if fine internal clearances, such as have been dealt with, are an advantage, then the common base for the lot—the bed plate—must be of a nature to ensure the clearance remaining concentric, and also to maintain true alignment with the operating motor. For these reasons deeper sections than are usual are an advantage. With the "ring type" of casing the bed plate plays a more important part than with the "cylindrical" type on account of the lesser intrinsic rigidity of the contiguous type, and the greater necessity for a plane true surface in erecting. Hence the importance of deeper sections becomes more pronounced still. It is now customary to have supporting feet on each "ring" of a multi-chamber pump, probably more for the convenience of re-erection in awkward situations than for any necessity for actual support. Chambers, when dismantling or re-erecting, require to be slipped along the spindle and bed, and for this reason it is usual to machine the latter for its full length. An example of substantial base-plate provision for a "cylindrical" casing is shown in Fig. 32, and a bed as recommended for a "ring" casing pump is shown diagrammatically in Fig. 33. In cases where there is a heavy torque from the driving motor, diagonal ribs, as shown, are most valuable in resisting twist on the bed.

Bearings.—The almost "standard" speed of 1,450 revolutions per minute for operation of turbine pumps does not require water-cooled bearings. For outside bearings good design oil-ring lubricated type, as shown in Fig. 34, are used, a recommended feature being the provision of a thrower or guard to prevent water from a leaking gland entering the oil well and washing out the oil. As has already been stated, to keep the distance between shaft supports as short as possible involves the use of internal grease-lubricated bearings, typical designs of which are shown in Fig. 35. This type of bearing, when carefully designed, can be made to operate very satisfactorily in practice, despite a contrary view held for some time by some constructors. The important point of arranging for similar hydraulic pressures at each end of an internal

TYPES OF CASING.

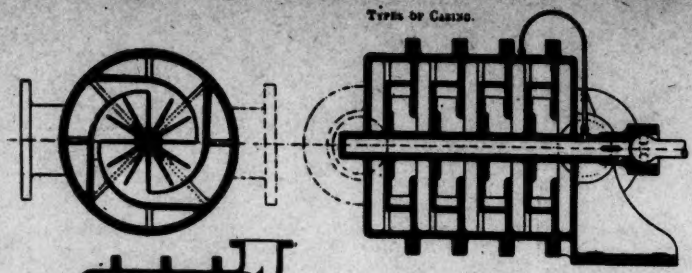


FIG. 1 a.
(Osborne Reynolds.)

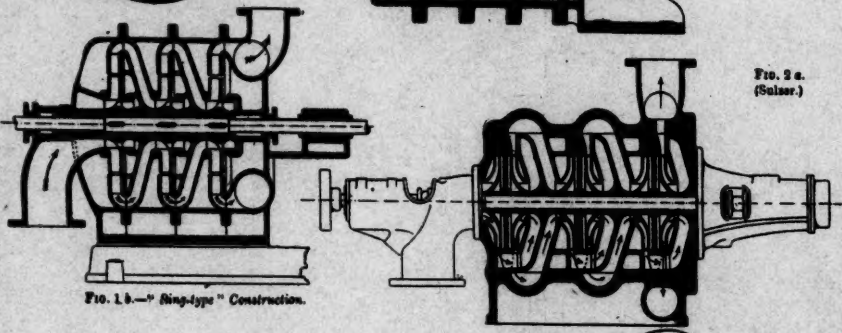


FIG. 2 a.
(Sulzer.)

FIG. 1 b.—"Ring-type" Construction.

FIG. 5.—Form of Guide-Passage. Flow

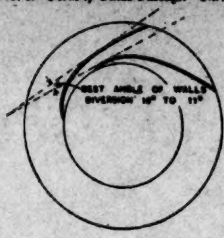


FIG. 7.—Flow tange
a. Flat.

TYPES OF CASING

FIG. 2 b.—Continental Pump.
Cylindrical-casing" Construction

FIG. 4.—Combination of
"Cylindrical" and "Ring" Feature.

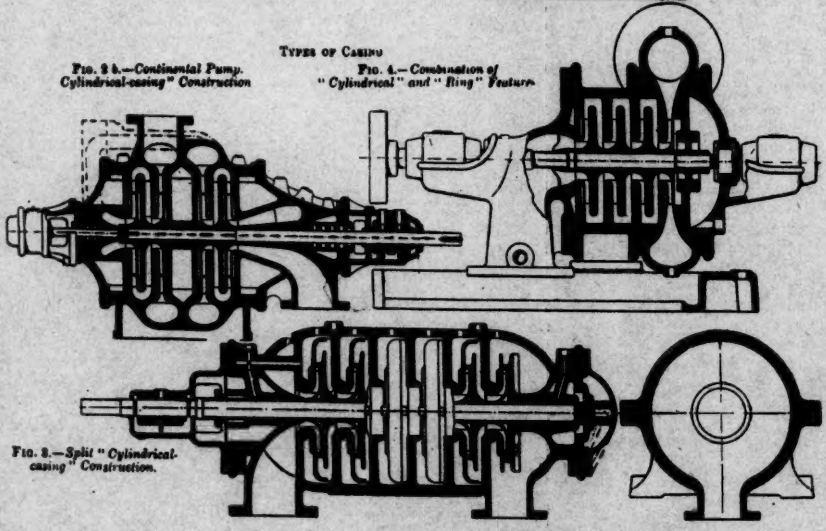


FIG. 3.—Split "Cylindrical-casing" Construction.

FIG. 9
(Marchand)

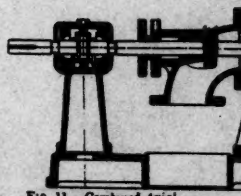


FIG. 11.—Combined Axial
and Spiral Flow

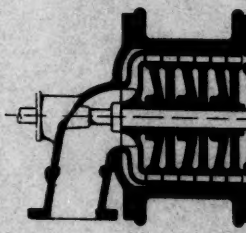
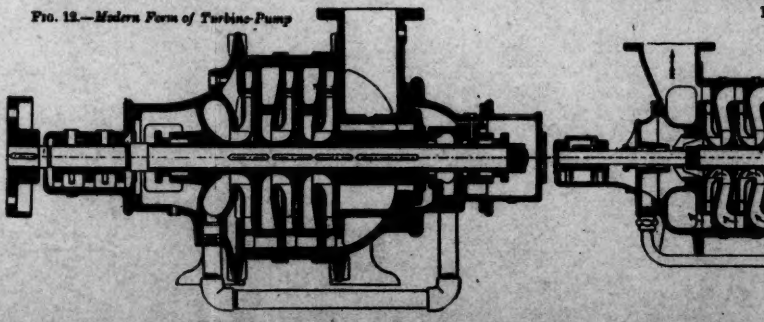


FIG. 12.—Modern Form of Turbine-Pump

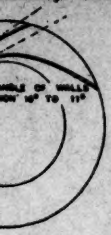


Types of Guide-Vanes.

Fig. 6.—"Stage" or "Cell." Flow tangential and radial, return radial.

Figs. 14 and 15.—Impellers.

Unequal side area.
One rubbing shoulder.



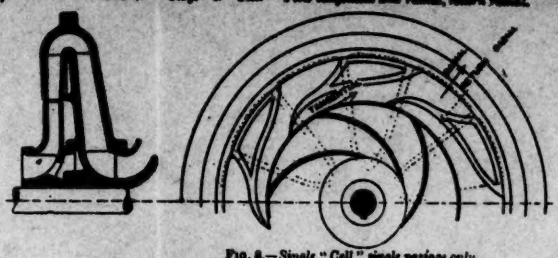
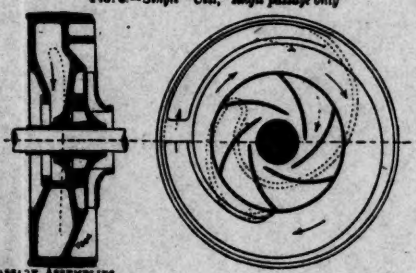


Fig. 8.—*Simple "Cell," single passage only*

Fig. 7.—*Flow tangential and spiral*
b. Perspective.



Combination Passage Assemblies

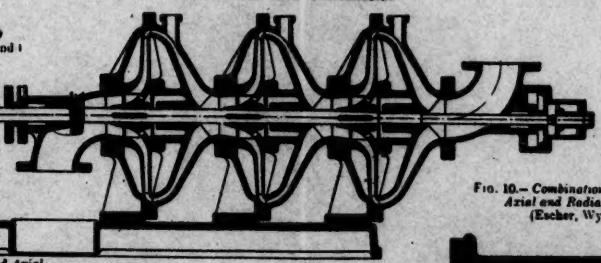
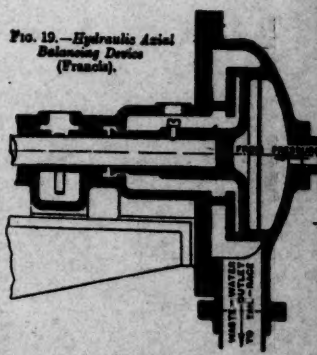
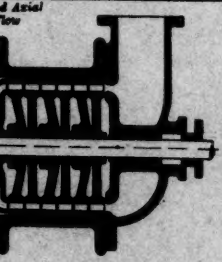


Fig. 10.—*Combination of Tangential, Axial and Radial Flow.*
(Escher, Wyss.)

Fig. 19.—*Hydraulic Axial Balancing Device (Francis).*





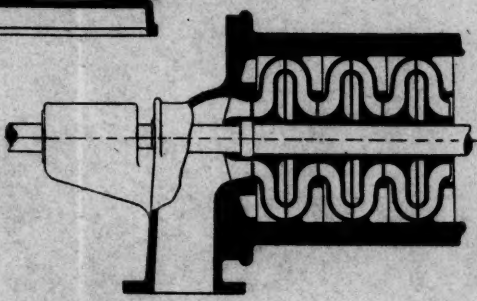


Fig. 13.—(Parcot.)

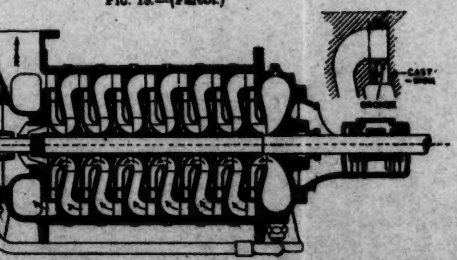
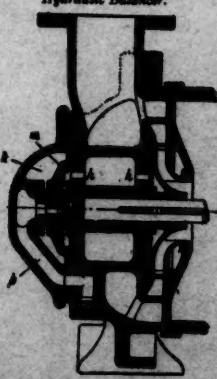


Fig. 23.—*Differential Hydraulic Balancer.*

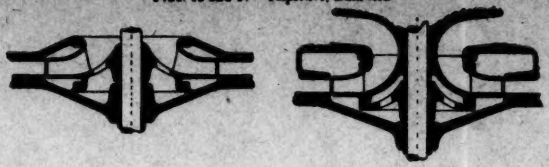


THE CONS

15.—*Impellers, Unbalanced.*

Equal side areas.
Two rubbing shoulders

Figs. 16 and 17.—*Impellers, Balanced*



Continental Hydraulic Bo

Fig. 27.—*Differentia*

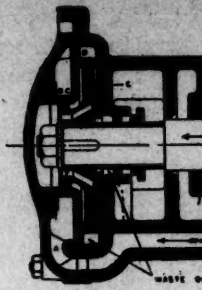


Fig. 18.—*Effects of Gyratory Pressures on End Thrust.*

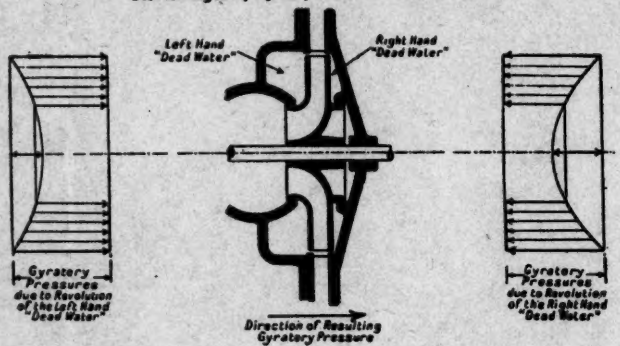

Left Hand "Dead Water"
Right Hand "Dead Water"
Gyratory Pressures due to Revolution of the Left Hand "Dead Water"
Gyratory Pressures due to Revolution of the Right Hand "Dead Water"
Direction of Resulting Gyratory Pressure

Fig. 28.—*Single-*

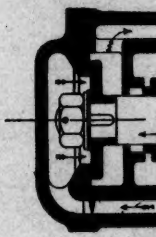


Fig. 20.—*Pump (Rateau) showing Hydraulic Compensator.*

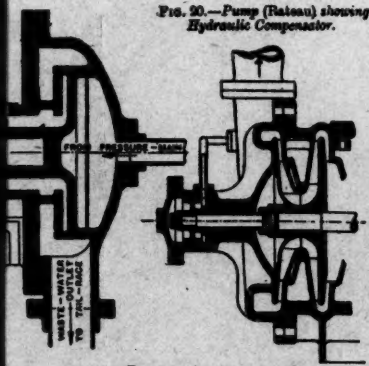


Fig. 21.—*Hand-adjusted Balancer.*

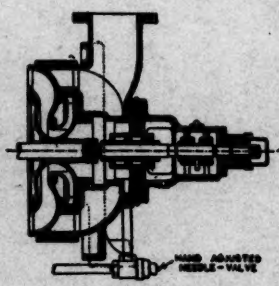


Fig. 22.—*Mechanically adjusted Balancer.*

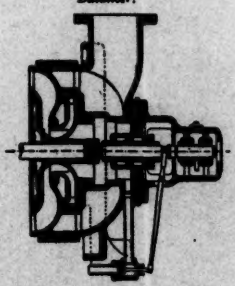


Fig. 24.—*Single-plate Balancer.* (Sulzer).

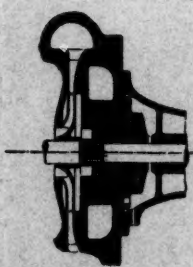


Fig. 31.—*Types of I*

Single-plate Balancers.

Fig. 25 (Millington.)

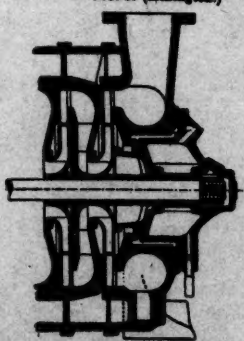
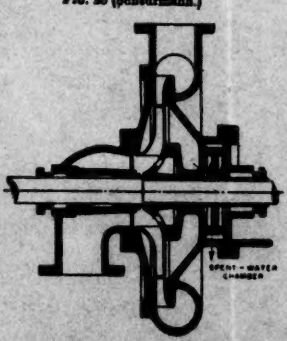


Fig. 26 (Schuurmann.)



CONSTRUCTION OF TURBINE PUMPS.

...dal Hydraulic Balancer.
...g. 27.—Differential.

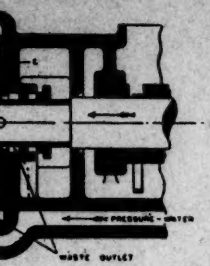
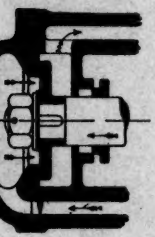


Fig. 28.—Single-plate.



...g. 31.—Types of Impellers.

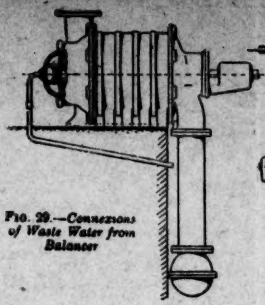


Fig. 29.—Connexions of Waste Water from Balancer

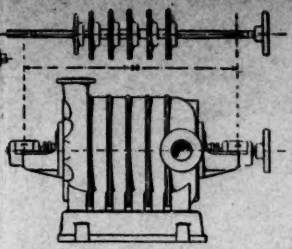


Fig 30 a

Span of Bearings—
Fig 30 b.

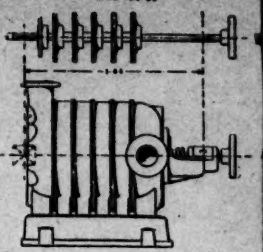


Fig. 32.—Bed for "Cylindrical" type of Casing.

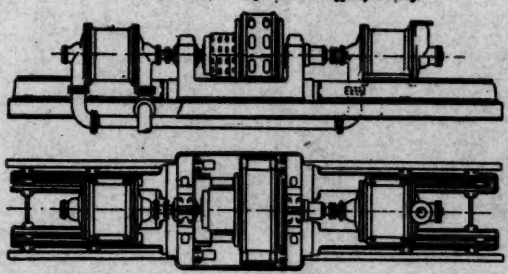


Fig. 33.—Baseplate for "Ring

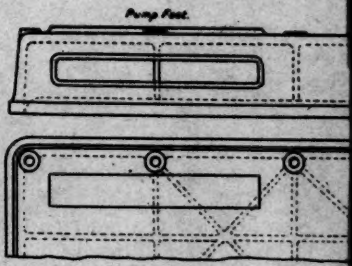


Fig. 35.—Internal Grease-Lubricated Bearings.

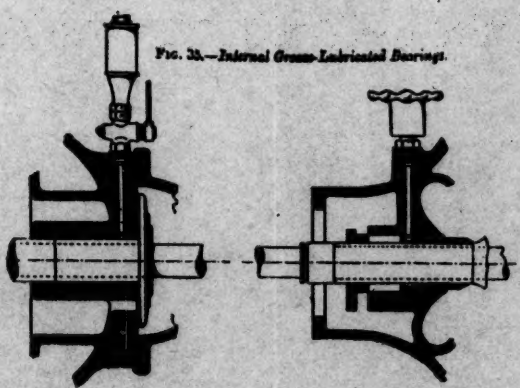
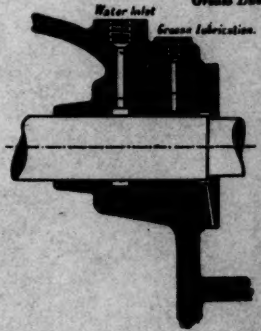


Fig. 36.—Hand-op
Grease Lubricat



IPS.

Continental Hydraulic Balancer.

Fig. 27.—*Differential.*

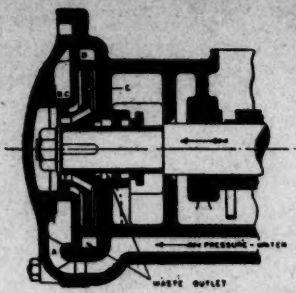
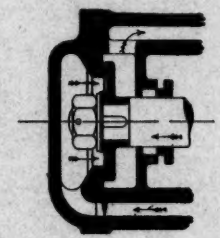


Fig. 28.—*Single-plate.*



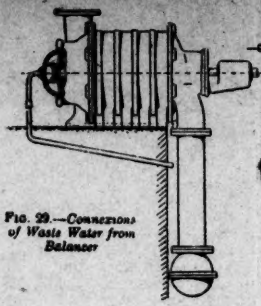


Fig. 29.—*Connexions of Waste Water from Balancer*

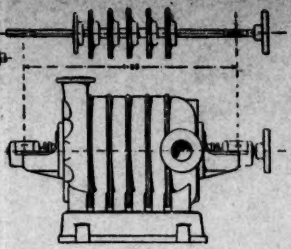
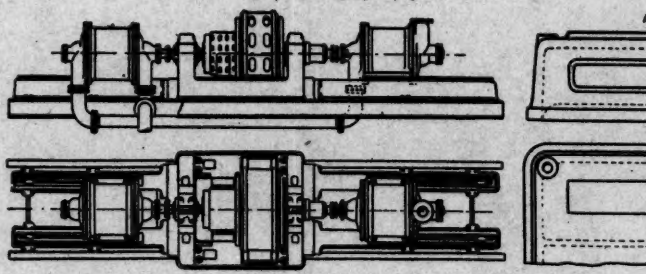


Fig. 30 a

Fig. 32.—*Bed for "Cylindrical" type of Casing.*



Mechanically adjusted Balancer.

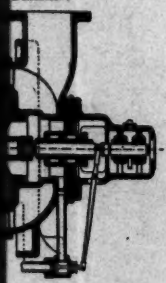


Fig. 31.—*Types of Impellers.*

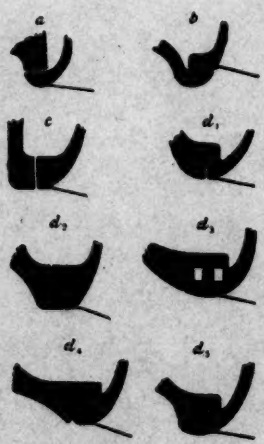
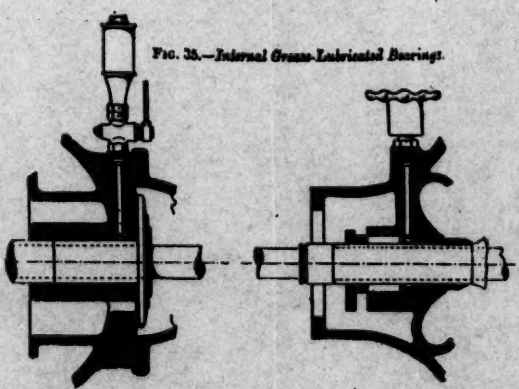
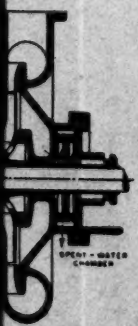


Fig. 33.—*Internal Grease-Lubricated Bearings.*





BINE PUMPS.

Span of Bearings—Three arrangements.

Fig 30 b.

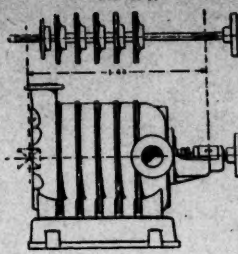


Fig 30 c.

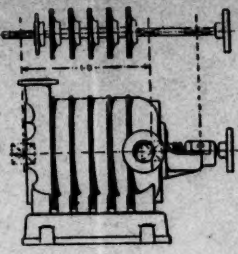


Fig. 34.—External Bearing with Oil-ring Lubrication

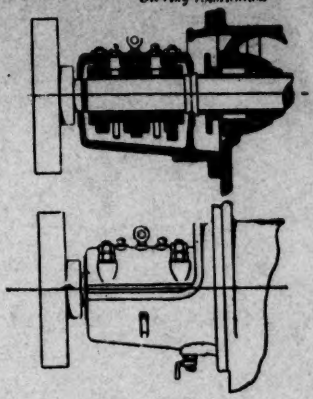


Fig. 33.—Baseplate for "Ring" type of Pump.

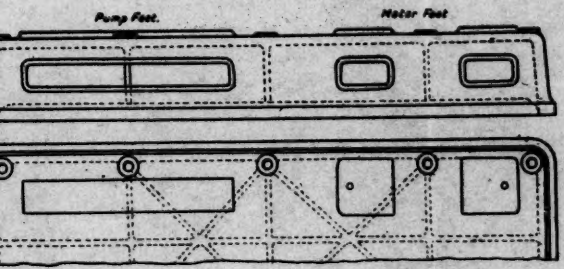


Fig. 37.—Suggestions for Special Gland Bearings.

Loose flange bearing on two rounded points only.

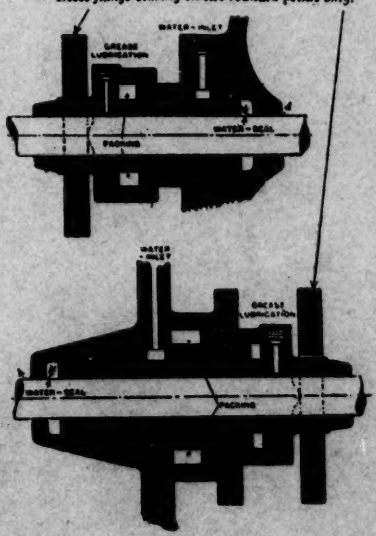
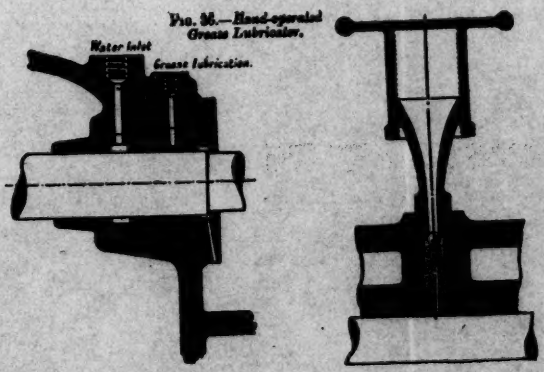


Fig. 36.—Hand-operated Grease Lubricator.



grease-lubricated bearing, so as to prevent any risk of flow through the bearing surfaces, must be rigidly followed for best results. The charging and supplying of grease to internal bearings requires proper lubricators. The ordinary "Stauffer" is unsuitable, though generally a hand-operated appliance of simple design—such as Fig. 36—meets the case. A long trumpet-shaped entrance is essential in order to force against the high pressure due to the head of water against which the pump is working. The selection of a good quality grease lubricant, suitable for operating in water, plays an important part in the satisfactory lubrication of a bearing.

Stuffing boxes.—It is doubtful if we can accept the customary stuffing box and gland as good and satisfactory practice. The functions of this detail are, on the suction and vacuum end, to prevent air leakage into the pump, and, on the delivery end, to prevent pressure leakage from the pump. The action of "packing" on rapidly-rotating shafts is so severe as to demand the omission of a stuffing box wherever possible. On the delivery end of a pump it is easy to avoid a gland, either by closing in the spindle—Figs. 3, 23, and 25—or by so reducing the pressure in passing through a hydraulic balancer that there is no pressure against which to pack—Figs. 4, 12, and 13. Some constructors have done without stuffing boxes at both ends of the pump for some time, and they generally rely on pressure water from the pump itself supplying the necessary packing to prevent access of air to the suction inlet. The possibility of difficulty in starting up a pump before the water packing is available should be carefully considered when this method is adopted.

Suggestions for the substitution of a special form of grease-lubricated gland bearing, without any packing in contact with the rotating spindle, are shown in Fig. 37. Though these bearings are not arranged with end pressure in equilibrium, there is no tendency for water to pass over the bearing surfaces, as leakage will take place from the water-sealing ring to the reduced pressure of the suction inlet.—"The Engineer."

THE DETERIORATION OF TURBINE BLADING.*

We reproduce below a highly important and extremely interested paper read before the South African Institution of Engineers by Mr. A. Fenwick. The paper describes various mishaps to the turbines at the power stations of the Rand power companies. The following particulars of the plant are taken from Garcke's "Manual of Electrical Undertakings":

Station.	*Plant.*	*Maker.*
	Kw. turbo-alternators.	
Brakpan	2–3000	A.E.G.
	2–11000	A.E.G.
Rosherville	5–9600	A.E.G.
Vereeniging	2–9600	A.E.G.
	2–12000	A.E.G.

In the discussion which followed the reading of the paper Mr. Bernard Price, the chief engineer to the power companies, stated that the defects enumerated in the author's paper had resulted in very heavy maintenance charges.

It is perhaps unnecessary to add that the extraordinary failures enumerated are not inherent in the impulse type of turbine, in which, as with others, excessive wear is merely the penalty for the use of improper materials or designs.

The text of Mr. Fenwick's paper is as follows:

*The original illustrations as they appeared in "Engineering" have been reduced about one-fourth.

THE DETERIORATION OF TURBINE BLADING.

Introduction.—The object of this paper is to place on record the behavior of certain materials used in turbine blading. The information and data given below summarize the experience gained during the past six years in operating and maintaining the turbine plant of the Rand power companies (V.F. and T.P. Company, Limited, and R.M.P.S. Company, Limited).

All necessary particulars regarding the design of this plant have been included, and it is hoped that the publication of this experience will be of value to other power companies and to manufacturing firms, and that the discussion on this paper will serve to further elucidate problems connected with turbine blading.

PARTICULARS OF TURBINES INSTALLED.

Reference will be made to various turbines, which, although operating on the same principle, differ in design. Table 1 gives particulars of the plant dealt with.

Fig. 1, page 542 shows the turbo shaft with wheels assembled, the inlet nozzle, stationary segment, and a portion of each guide-blade diaphragm in position for the turbines under Class 4.

In all classes the steam passes through the inlet nozzles and impinges on the first row of Curtis moving blades. These nozzles are of cast iron, and are made in sections containing five to eight ports; one section only is used for turbines under Classes 1, 3, 6 and 7, two sections for Classes 2 and 4, and three sections for Class 5. In each case the nozzles are opposite to a portion only of the wheel blading, the length of arc varying from approximately one-sixth to one-third of the whole circumference, so that the admission is partial. After passing through the first row of Curtis moving blades the steam passes through a row of stationary blades fixed to a steel segment, which is bolted to the turbine casing. This Curtis stationary segment also embraces a portion only of the wheel circumference, the arc being slightly longer than that covered by the inlet nozzles. The stationary segment blades are somewhat similar in size and shape to the Curtis moving blades, and direct the steam to each successive row of blades in the Curtis wheel, so that there is one row of blades less in the segment than there are rows in the Curtis wheel. After leaving the last row of Curtis wheel blades the steam discharges into the casing in which the Curtis wheel revolves, the pressure in this casing varying from two to six atmospheres, according to the type of machine and the load which it is carrying.

The Curtis casing is a closed compartment separated from the Rateau portion by a cast-iron diaphragm; on the outer periphery of this diaphragm is a ring of guide blades forming nozzles, which convert the steam pressure into velocity and give the steam the correct angle for impinging on the moving blades. The admission in this case is complete, the nozzles extending round the entire circumference of the diaphragm. The first ring of guide blades directs the steam from the Curtis casing on to the first Rateau wheel, or, as it is termed in this paper, the second stage wheel. This process is continued throughout the turbine, so that each wheel is in its own cell or compartment and divided from the next by a cast-iron diaphragm and ring of guide blades.

Two main types of guide blading are in use; the first—termed a guide-blade ring—consists of an inner and outer retaining ring cast round the blades. Fig. 2 shows one of these rings, which was made locally; the outer portion fits into the turbine casing, and the inner portion supports a cast-iron undivided diaphragm which partitions off one stage from the

TABLE I.—PARTICULARS OF TURBINES INSTALLED.

Class.	Output.	Installed at	Date.	No. of sets.	Steam pressure stop valve.	Steam temperature stop valve.	Steam temperature Curtis casing.	Temperature vacuum.	Revs. per Minute.	Mean blade velocity.
					lbs. per sq. in.	deg. C.	deg. C.	deg. C.		ft. per sec.
1	3,000-kw. generator turbines.	Simmer Pan..	1909-10	6	200	300-310	220	35-40	1,500	Curtis 466
		Brakpan.....	1908-9	2	190	280-290	200	40-45	1,500	Rateau 410
2	9,600-kw. generator turbines.	Rosherville...	1911	5	220	320	240-250	40-45	1,000	325 to 350
		Vereeniging..	1912	4	220	350	250-260	40-45	1,000	
3	4,000-h.p. compressor turbines.	Rosherville...	1911 1913	6	220	320	220	35-40	3,000	Curtis 515 Rateau 645
4	12,000-kw. generator turbines.	Simmer Pan..	1914	2	220	350	250-260	35-45	1,500	Curtis 516 St. 2 to 11
		Brakpan.....	1914-15	2	220	350	250-260	35-45	1,500	456 to 392 St. 12 582
5	10,000-h.p. compressor turbines.	Rosherville...	1914	3	220	330	220	35-40	3,000	Curtis 618 Rateau 721
6	300-h.p. auxiliary turbines.	Rosherville...	1911	11	220	320	245	...	1,400	Curtis 200
		Vereeniging..	1912	4	220	350	255	...	1,400	
7	550-h.p. circulating pump turbines.	Vereeniging..	1912	4	220	350	250	...	1,040	Curtis 230
		Simmer Pan..	1914	2	220	350	250	...	1,040	

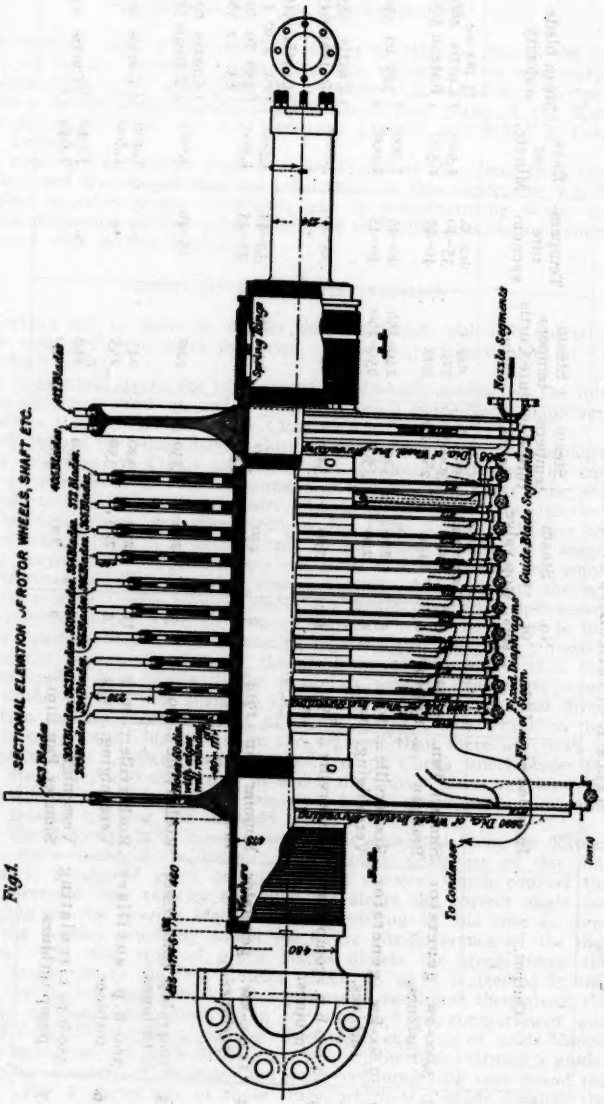

SECTIONAL ELEVATION OF ROTOR WHEELS, SHAFT ETC.
457 Blades
Spring Stop
Nozzle Segments
Flow of Steam
Guide Blade Segments
To Condenser

Fig. 1.

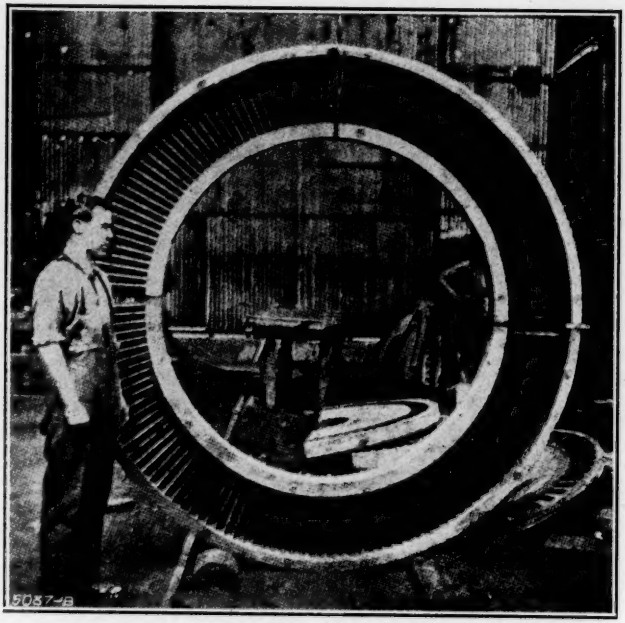


FIG. 2.

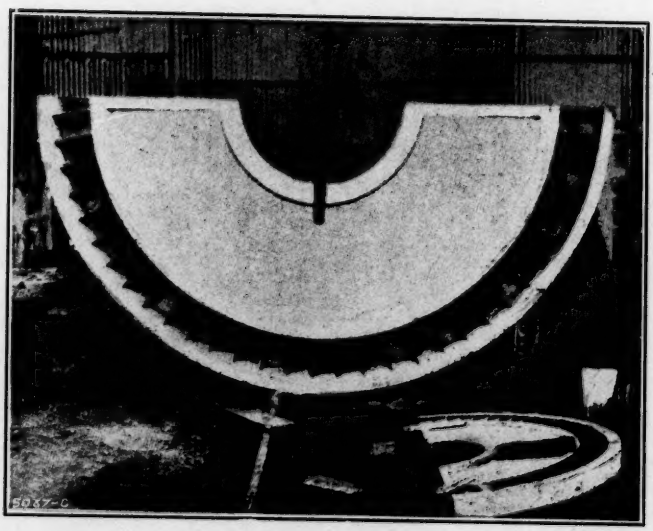


FIG. 3.

next. This type is in use on turbines Class 1 and 2. The guide ring is in two halves, which butt together at the horizontal joint of the turbine casing.

In the second type the diaphragm forms an integral part with the retaining rings, the complete diaphragm being cast in two halves, one of which is shown in Fig. 3. To avoid confusion this is called a guide-blade diaphragm, and is used on turbines under Classes 3, 4 and 5. This paper does not deal with the merits of either type; where failure of the blades occur, however, the results are usually totally different, and this matter will be referred to later.

In the case of the 9,600-kw. turbines (Class 2), the diaphragms are undivided and have to be lifted into, or removed from, the turbine casing with the rotor. These diaphragms rest on the lower half of each guide ring, the rings being bolted to the turbine casing; the upper portion of the turbine casing carries the other halves of the guide rings, and a tongue on the latter fits into a corresponding groove in the diaphragm to prevent lateral movement. The weight of this turbine rotor complete is 26 tons.

TABLE II.—SHOWING VARIOUS METALS IN USE FOR BLADING.

Mark.	Material.	Class of turbine.	Wheel or guide blades.
A	4 per cent. nickel steel.	2 and 3.....	Guide.
B	Siemens-Martin steel.....	2, 3, 4 and 5.....	Guide.
C	5 per cent. nickel steel...	2, 3, 4 and 5.....	Guide.
A1	3 per cent. nickel steel...	3.....	Guide.
H	Medium carbon steel.....	2, 4 and 5.....	Guide.
J	30 per cent. to 32 per cent. nickel steel.	1, 2, 3, 4 and 5...	Guide.
K	25 per cent. nickel steel..	3.....	Wheel and Curtis segment.
P	5 per cent. nickel steel..	All classes	Wheel and Curtis stationary segment.
L	Aluminum bronze.....	1, 2, 3, 4, 5 and 6.	Wheel and Curtis stationary segment.
M	Brass.....	All classes.....	Wheel and Curtis stationary segment.
N	Nickel brass.....	3.....	Wheel.
O	Monel metal.....	3.....	Wheel and Curtis stationary segment.

MATERIALS USED.

Table II shows the metals in present use on the various turbines. Some of these metals have been in use for a few months only, and are, therefore, in the experimental stage. Tables III and IV give the analyses of steels and non-ferrous alloys in use. Table V shows the materials in use for wheel blading.

Referring to Table II, it will be seen that materials A, B, C, A1, H and J are used for guide blading only. Of these, A, B, C and A1 have been used recently for locally manufactured guide blading, and have only been in commission for a few months; information as to their suitability for this particular purpose is therefore not available. The analyses given are of plates from which the blades are made.

Material H, as far as the author is aware, is only in use on turbines

TABLE III.—ANALYSES OF STEELS IN USE.

Mark.	C.	P.	S.	Si.	Mn.	Ni.	W. (Tungsten.)	Cr. (Chromium.)	Fe.	Remarks.
	per cent.	per cent.								
A	0.200	0.026	0.022	0.198	0.767	3.980	0.128	0.108	94.571	For guide blading only. Experimental stage.
B	0.400	0.058	0.050	0.035	0.936	.....	.....	.....	98.521	For guide blading only. Experimental stage.
C	0.286	0.056	0.030	0.120	0.648	4.860	.....	.....	94.000	For guide blading only. Experimental stage.
A1	0.273	0.045	0.028	0.046	0.612	3.170	.....	.....	95.826	For guide blading only. Experimental stage.
H	0.350	0.030	0.040	0.200	0.500	.....	.....	.....	98.880	Guide blading only. Considerable pitting in some cases.
J	0.310	0.036	0.046	0.350	0.540	30.650	.....	.....	68.068	Very unsatisfactory. Cracks badly.
K	.....	.....	.....	.....	.....	25 per cent.	.....	.....	.....	Analyzed for nickel only. Unsatisfactory. Used for wheels and Curtis stationary segments.
P	0.204	0.016	0.010	0.046	0.288	4.750	0.350	.....	94.336	For wheel blading only.

under Classes 4 and 5, and on one turbine only under Class 2 (No. 4, Rosherville).

Material J is the steel principally used for guide blading, and occurs in some stages on every one of the turbines dealt with, excepting the auxiliary turbines, Classes 6 and 7, which have no guide blading. On turbines Classes 1, 2 and 3 it has been used exclusively, with the exception of No. 4, Rosherville.

TABLE IV.—ANALYSES OF NON-FERROUS ALLOYS IN USE (USED FOR WHEEL BLADING ONLY).

Mark.	Cu.	Al.	Fe.	Zn.	Pb.	Ni.	Tensile Strength of Tons per Sq. In.	Specific Gravity.	Remarks.
L	89.00	8.5	2.5	—	—	—	44 actual blade..	7.73	Aluminium bronze.—Very unsatisfactory in high temperature stages.
M	72.50	—	0.01	27.48	Trace	—	—	8.58	Brass.—Stands well.
N	50.00	—	0.6	39.4	—	10.00	37 actual blade..	8.43	Nickel brass.—Experimental
O	30.00	—	1.50	—	—	68.50	Approx. 40 rolled bar	—	Monel metal.—Experimental

TABLE V.—MATERIALS USED FOR WHEEL BLADING OF ALL TURBINES.

Class of Turbine.	Curtis Stage. Row. 1.	2.	3.	4.	Rateau Stages. Row. 2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.	13.	Remarks.
1–3,000-kw.	L	L	—	—	L	L	L	L	L	L	L	L	—	—	—	—	On Sets 5 and 6 Simmer Pan stages 1 to 4 are bladed with P steel.
2–9,600-kw.	†	P	L	—	P	P	P	P	M	M	M	M	L	L	L	L	Stage 2 is not fitted on sets 3 and 4, Vereeniging, which are rated at 12,000 kw. These two sets are bladed with P steel for the third row Curtis stage.
					Practically no deterioration on these stages. Maximum running hours, 30,673.												
3–4,000-h.p.	P*	L	—	—	L	L	L	L	†—	—	—	—	—	—	—	—	* One set only was fitted with K steel on Row 1.
																	† L bronze now replaced with P steel on all sets for stage 6.
4–12,000-kw.—																	
Simmer Pan	P	P	—	—	P	M	M	M	M	M	L	L	P	P	P	—	NOTE.—P steel is used for stages 3 and 4 in Brakpan sets, and M brass for same stages in Simmer Pan sets.
Brakpan	P	P	—	—	P	P	P	M	M	M	L	L	P	P	P	—	
5–10,000-h.p.	P	P	—	—	M	M	L	P*	—	—	—	—	—	—	—	—	† There are two wheels in parallel for stage 5, both bladed with P steel.
6–300-h.p.—																	
Rosherville	P	P	L	L													
Vereeniging	P	P	M	M													
7–550-h.p.—																	
Vereeniging	P	M	M	M													

P = 5 per cent. nickel steel. L = Aluminium bronze. M = brass.

Analyses of the steel plate from which the blades H and J were manufactured are not available, but several analyses of blades in both good and bad condition have been carried out, showing the J steel to contain approximately 30 per cent. to 32 per cent. nickel. All discarded blades have not been analyzed, neither have all those at present in commission; blades containing no nickel, however, have the distinguishing mark H, and this has only been found on certain stages in generator turbine No. 4, Rosherville, and turbines Classes 4 and 5.

For wheel blading P, L and M are in general use, and ample evidence of the behavior of these alloys is to hand. N and O have been used quite recently for wheel and stationary segment blading, but sufficient time has not elapsed to form an opinion of their suitability for this purpose.

WHEEL BLADING.

An examination of these machines shows a variation of some thousands of running hours in the life of blades of similar material and position in the turbine, and the life here represents the running period up to the time reblading was carried out; in some cases the blades could safely have been run for a longer period, but, owing perhaps to a com-

plete dismantling of the set or extensive repairs to later stages, it has been advisable to reblade. In other cases, owing to difficulties in taking the set out of commission, blades have run much longer than they should have done, considering the safety of the machine.

It will be noticed that in some cases, where the wheel and guide blading have not been renewed, the running hours differ for the same turbine. This is explained by the fact that the complete turbo drum was removed for repairs, a spare drum taking its place.

3,000-*kw. Turbines, Class* 1.—Dealing first with the Curtis stage, and the first three low-pressure stages, two materials have been tried, viz., aluminum bronze (L) and 5 per cent. nickel steel (P), and the superiority of the latter is apparent. No particular trouble has been experienced from corrosion of the steel, although if the machine is standing any length of time the blades become rusty, which, however, soon wears off with use. Sets 5 and 6, Simmer Pan, were fitted with steel blades, and have been running for six years. In neither case has failure occurred, the running hours up to December 31, 1916, being 39,119 and 39,298 respectively.

The aluminum bronze blades have a very different record, and are quite unsuitable for this part of the turbine. The aluminum content disappears from the surface of the metal, leaving a very brittle layer of copper; where the section of metal is thin, as at the inlet and discharge edge of the blades, it breaks away like glass with light tapping. With continued use the outer layer cracks and flakes off, exposing the part underneath, which in turn flakes off until the blade is split up into several layers. Where the section of the blade is fairly heavy, as in the Curtis stage, the shape is gradually lost, but the body of the blade remains sound and seldom fails. What usually happens is, the portion which is riveted over to secure the shrouding chips off, allowing the latter to come adrift.

Figs. 4 and 5 (specimen 6/13) show a good example of this deterioration, and are from the first row of Curtis moving blades, after being in commission for about 24,000 hours. The flaking of the metal referred to is very much in evidence, and the rivet heads in this sample are particularly bad, having practically disappeared.

Fig. 6 (specimen 7/13) is from the second row of Curtis blades on the same machine; the deterioration is similar, but not so far advanced; the running hours are the same. The running time before reblading was carried out varied considerably, but an average has been taken for the first four stages on sets 1 to 4, Simmer Pan, and works out at 26,700 hours.

The Curtis blades have a shorter life than the later stages, and this is no doubt due to the higher temperature in the Curtis casing. After leaving stage 4 the deterioration takes a different form, erosion or pitting of the back or convex portion of the blade being usual. The inlet and discharge edges of the blades also become very thin and holed, and are easily broken off. Fig. 7 (specimens 13, 10, 11, 12/13) shows typical examples. The blades are from stages 5, 6, 7 and 8 respectively, on No. 5 set, Simmer Pan, and in this case ran for 33,665 hours before being renewed. While a certain amount of erosion has taken place, due to minute particles of matter passing through with the steam and a certain amount of moisture in the latter, the pitting looks as though chemical action had taken place, and may be due to certain impurities in the steam becoming active in the presence of moisture. It is curious that the action should be confined to these stages, as the two following stages (9 and 10), also of aluminum bronze, do not suffer. The four examples here shown would undoubtedly have lasted several thousands of hours longer before actual failure occurred. It was thought possible,

Fig. 4

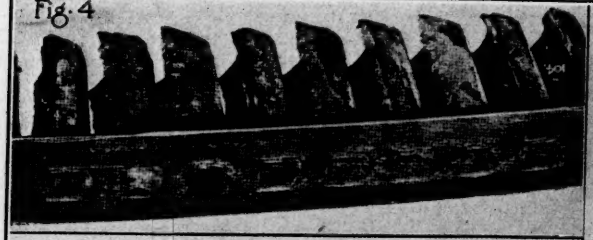


Fig. 5

Fig. 6

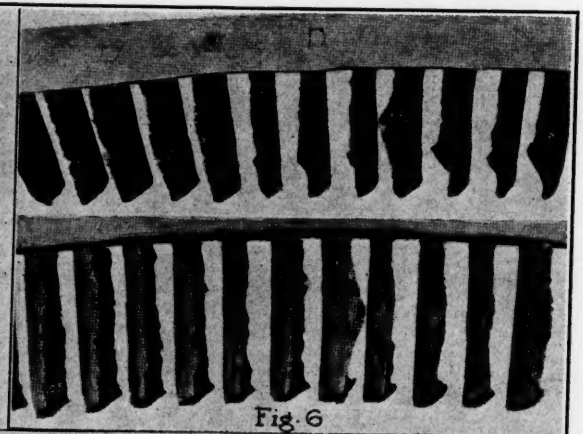


Fig. 7

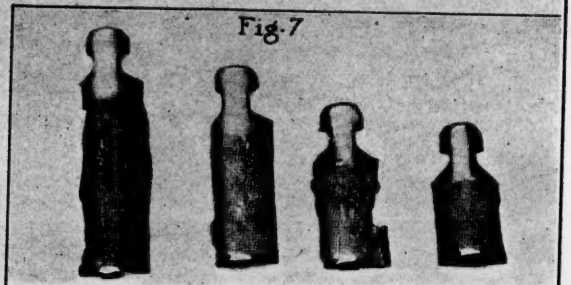


Fig. 8

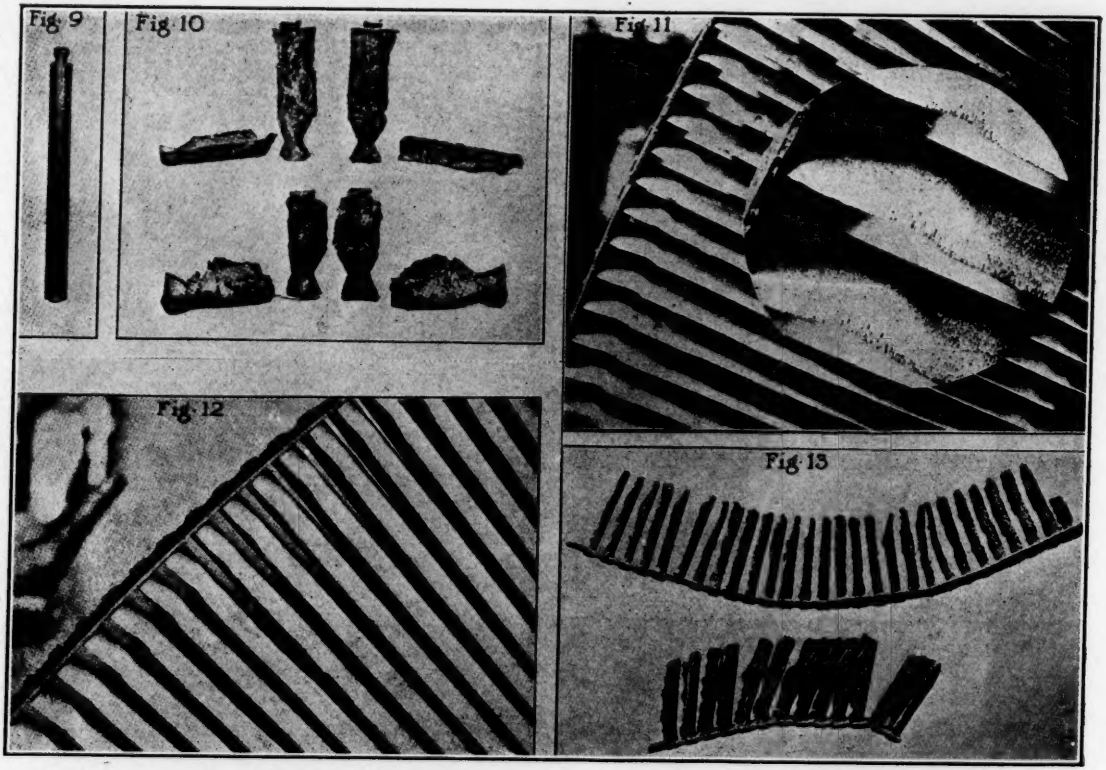


Fig. 9
Fig. 10
Fig. 11
Fig. 12
Fig. 13

BRASS.—MAGNIFIED 60 TIMES.

COMPOSITION:—Cu 72.50, Fe 0.01, Zn 27.48, Pb Trace; Specific Gravity 8.58.

ALUMINUM BRONZE. MAGNIFIED 60 TIMES.

COMPOSITION:—Cu 89.00, Fe 2.5, Al 8.5; Specific Gravity 7.73.

"H" STEEL. MAGNIFIED 100 TIMES.

COMPOSITION:—C 0.350, P 0.030, S 0.040, Si 0.200, Mn 0.5, Fe 98.880.

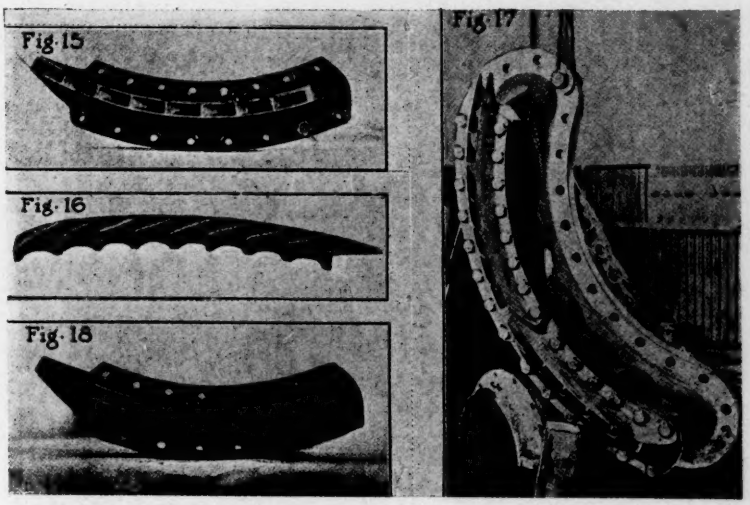


Fig. 19

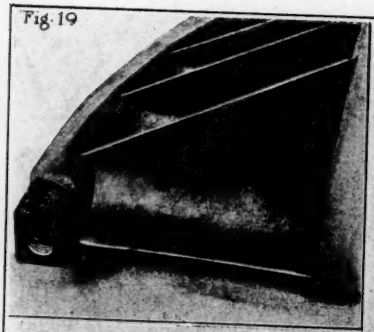


Fig. 20

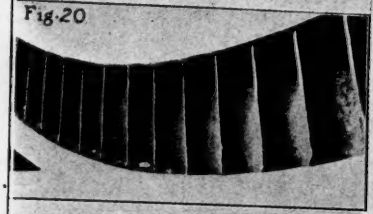
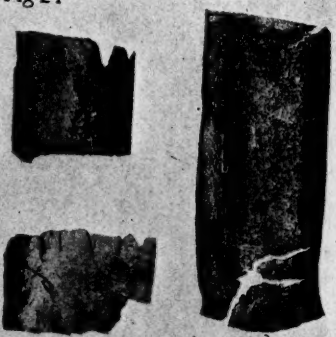


Fig. 21



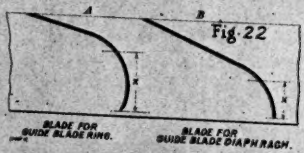


Fig. 23

Fig. 24

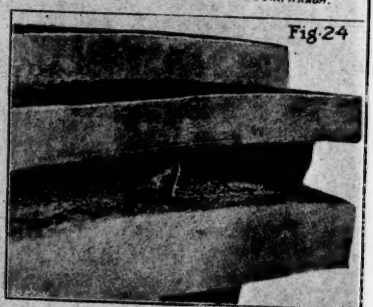
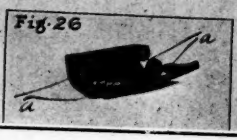


Fig. 25

Fig. 26



however, that the roughness of the blades would detract from the efficiency of the set. After reblading, careful tests showed that no appreciable improvement had been made in steam consumption. The last two stages stand up well, the present running hours varying from 39,000 to 46,500 on the first four sets at Simmer Pan, and up to 52,000 on one of the two sets at Brakpan, which is in each case the total running time of the machine. In no case have these stages had to be renewed owing to deterioration, although the edges are naturally worn somewhat, and in many cases have suffered mechanical damage by fouling broken pieces of guide blading, and had to be restraightened.

Our present practice is to fit the first four stages with 5 per cent. nickel steel blades, stages 5 to 8 with brass blades, and the last two stages with aluminum bronze. Brass has not been in commission long enough on these particular sets to show whether it will prove more satisfactory than the bronze, though, judging by experience obtained on other types of machines, it is highly probable. It may also be found that brass would do equally as well as bronze in the last two stages, and, personally, the author is of opinion that 5 per cent. nickel steel for the first eight stages and aluminum bronze for the last two would prove very durable and give good results.

96,000-kw. Turbines, Class 2.—The blading has, with two exceptions, proved satisfactory up to the present, and Table V, giving particulars of wheel blading, shows the material originally used. The maximum running hours for the set which has been longest in commission are 30,673 to December 31st, 1916, with the exception of row 3, Curtis stage, the running hours of which vary from 15,000 to 23,490. The other turbines under this class have at present run less than 30,000 hours, the time varying with the dates of installation. With the exception of row 3, Curtis stage, and stage 13, referred to later, none of the blading has been renewed unless damaged mechanically and shows no visible deterioration of the metal; on the first row of Curtis blades the entrance edge is worn slightly concave, not sufficient, however, to warrant renewal. On the non-ferrous stages the deposit which collects under the blade shrouding has been found to contain copper, showing that a certain amount of erosion has occurred; this, however, is not visible on the blades themselves.

On row 3 (Curtis) the aluminum bronze proved distinctly unsatisfactory, and became so brittle that in some cases the blades broke away from the wheel, and in all cases the edges were so brittle within the running hours above mentioned that pieces could be knocked out with a slight tap. The bronze in this row has since been changed for steel, with satisfactory results. On sets 3 and 4, Vereeniging, steel was used from the first, and has run 25,000 hours, being at present in excellent condition.

Fig. 8 (specimens, 5, 6/15) shows good examples of aluminum bronze blades from row 3 (Curtis) from two different turbines; the blades were renewed after 15,000 and 18,000 hours respectively.

The brass blades in stages 6 to 9 are in excellent condition from a deterioration point of view, but in several cases have been severely damaged mechanically from various causes, and successfully straightened up.

The aluminum bronze blades in stages 10, 11 and 12 have proved satisfactory, but in stage 13 breakages of the blades have occurred The blades are secured to the wheel by a tee-headed root, and fracture frequently occurs across the neck of the tee. The weight of the complete blade is 9 ounces, and the mean velocity 327 feet per second, at a radius of 3.1 feet. The stress is, therefore, not excessive, being 601 pounds, which, taken over an area of 0.1 square inch, equals approximately 3 tons per square inch. It is very probable that vibration of the blade is largely

responsible for the failure. A tensile test was carried out at the Government laboratory on a new blade, and on one of the broken ones, showing the new material to stand 44.9 tons and the old 29.8 tons per square inch before fracture. It would seem, therefore, that the material has lost a good deal of its original strength, although there is very little difference in outward appearance. In some cases the blades have fractured about 1¼ inches above the rim of the wheel, *i.e.*, just above the distance piece.

Fig. 9 shows one of these blades after removal from the wheel, the crack across the blade being clearly visible. Some of these blades (stage 13) have run from 20,000 to 30,000 hours, and are still sound. Breakage occurred in three cases after 24,900 hours, 23,800 hours, and 10,600 hours respectively. It should be noted that erosion of the inlet edge of the blades in the later stages, due to wet steam, is almost invariably absent on the turbines just described; one or two isolated cases are on record, but are very slight.

4,000-H.P. Compressor Turbines, Class 3.—It will be noticed that 25 per cent. nickel steel has been used for row 1 of the Curtis stage; this, however, is an isolated case, as far as the author is aware, for moving blades on these turbines, although this material is in use for Curtis stationary blading. A peculiarity of this steel is its non-magnetic property.

In this particular case failure occurred after running for 17,700 hours, although in outward appearance the metal appears to be quite good. Specimen 7/11 showed one of these blades considerably battered; cracks were noticed near the root of the blade, showing the brittle nature of the material. An examination of the turbine after breakdown occurred showed the inlet nozzles to be intact, and although it is possible that some foreign matter may have passed through the nozzles and fouled the blades, it is improbable, and it may be concluded that these particular blades broke owing to brittleness of the metal acquired by use with high temperature steam and vibration.

Five per cent. nickel steel in use on the other sets for row 1 has given satisfaction up to the present, but the running hours to date are much less than the generator turbines previously referred to, 15,600 hours being the maximum.

In the second row of Curtis blades two materials have also been tried, namely, aluminum bronze (L) and 5 per cent. nickel steel (P). The former proved quite unsuitable in all cases, and has now been replaced by 5 per cent. nickel steel. The 5 per cent. nickel steel now in use for row 2 has so far proved satisfactory, but has only been tested for 11,500 running hours.

The aluminum bronze blades in stages 2, 3 and 4 have a similar record to those referred to above, and Fig. 10 (specimens 1/11 and 3/11) shows good examples from the second and fourth stages after 10,500 running hours. They certainly look as though the inside of the turbine was no place for them, although, judging by the quality of the metal in the heavy part of the blade, there is no doubt they would continue to do their work inefficiently for quite a long time. These three stages have now been replaced by brass, and although sufficient time has not elapsed to say that this material applied to these stages is satisfactory, examination after 11,000 hours does not show any deterioration, and it will certainly prove to be much more suitable than the aluminum bronze. The metal is quite soft and pliable, and the thin edges stand up remarkably well to the steam temperature.

In stage 5 aluminum bronze is again used, and also stands well. In no case has this stage been renewed owing to deterioration, the maximum running hours being 18,000 odd. The temperature is fairly low at this part of the machine, and this doubtless accounts for the longer life.

Although corrosion of the metal has not occurred, a good deal of wear has taken place, the edges of the blades being now very thin and fragile; this, however, is fair wear and tear, and cannot be avoided unless a much harder material is used.

Aluminum bronze was also tried in the sixth stage, but was unsuitable for mechanical reasons. After being in use from 5,000 to 7,000 hours the blades broke near the root just above the distance pieces, and it is extremely probable that vibration was the cause. Five per cent. nickel steel was then tried, but fractured in the same manner. Ultimately a 5 per cent. nickel steel blade was used with a reinforced root, which gave somewhat better results; the latter blades are, however, not free from breakage, and a material is required which is not so easily affected by vibration. "Monel' metal is also being given a trial, but has not run long enough to form an opinion as to its merits. Nickel brass (N) is also being experimented with for stage 5; this alloy has less tensile strength than aluminum bronze, being in the neighborhood of 36 tons per square inch. It may, however, possess other valuable properties which will render it useful for turbine work. The steel blades for stage 6, just referred to, show a small amount of erosion on the inlet edge of the blades after 6,000 hours.

Turbines, Classes 4 and 5.—These turbines have not been in commission for more than two years; the information therefore covers a much shorter period than on those previously described. Three metals are in use, 5 per cent. nickel steel, brass and bronze, arranged in the order given, starting from the H.P. end of the turbine. Steel is also used for the last three wheels in one case, and for the last two wheels in the other case, at the exhaust end. The metals appear to be in the correct position for the best results, and up to the present no trouble has been experienced.

One point should, however, be mentioned—erosion of the inlet edge of the steel blades at the exhaust end of the machine is noticeable in all cases after 5,000 to 6,000 running hours; in the case of the compressor turbines (Class 5) running at 3,000 r.p.m., with a blade velocity of 720 feet per second, the erosion is very pronounced, the edge of the blade being like a fine saw. An actual blade is not available, but the photos, Figs. 11 and 12, show this fairly clearly.

The above covers the whole of the main turbines installed. A short reference will now be made to two types of auxiliary turbines used for driving condensing sets.

Auxiliary Turbines, Class 6.—These turbines have a Curtis wheel only, with four rows of blades, and no low-pressure or Rateau wheels.

There are 15 turbines of this class in use, and in the earlier sets installed at Rosherville the first two rows of blades were of 5 per cent. nickel steel, which has given good results, and the last two rows were of aluminum bronze, the use of which for these small turbines, where the blade section is small, proved disastrous.

Corrosion was well advanced after 8,000 hours, and the blades frequently disappeared from the wheel after from 12,000 to 16,000 hours, usually at some inopportune moment, causing a good deal of inconvenience. Fig. 13 (specimens 2 and 4/9) shows typical examples of these blades, which would hardly be recognized as such after 15,000 hours and 18,000 hours respectively.

On the later sets installed at Vereeniging brass has been used for the last two years, and in one instance for the second row, and this material has given good results, being in good condition after 26,000 running hours.

Auxiliary Pump Turbines, Class 7.—This type of auxiliary has also a single Curtis wheel, but is somewhat larger than the previous one, and

is direct-coupled to a vertical pump for supplying condenser cooling water. Six of these are installed, and in no case has the blade material shown deterioration, both steel and brass being in excellent condition after a maximum running period of 26,000 hours.

General.—From the foregoing remarks it is perfectly clear that aluminum bronze for use as a blade material in the high-temperature stages of a turbine is totally unsuitable; it cannot be wholly condemned, however, as excellent results have been obtained when used in the low-temperature stages, and while brass might do as well in this position, one would not expect as long a life from brass owing to the soft nature of the material, the bronze being much harder and naturally more able to stand up to fair wear and tear.

The aluminum bronze in use is, as far as we are aware, of uniform composition. A few samples have been tested and the analysis is given in Table IV. A photomicrograph of the metal polished and etched is shown, in Fig. 14. The specimen has been prepared from a new blade cut across at right angles to the length.

Five per cent. nickel steel has so far given good service where used, and, as previously mentioned, in some of the later turbines has been put into the last two wheels at the exhaust end. Examination shows that a good deal of wet steam or water erosion is taking place in some cases, particularly on the steam inlet edge of the blade.

As our experience does not extend beyond 10,000 hours on these new sets, we are somewhat in the dark as to the future behavior of the steel blades in these low-temperature stages, but for the early stages up to, say, stage 5 or 6, there is no doubt that 5 per cent. nickel steel is an excellent material. The brass in use stands the temperature well, and in several cases has replaced aluminum bronze, with satisfactory results up to the present.

Our experience with brass blades covers a maximum period of just over 30,000 running hours, and in no case, so far, has renewal been necessary owing to deterioration. It is possible that this material will outlast the steel, as a certain amount of oxidation of the latter is bound to occur. A photomicrograph of the brass in use is shown in Fig. 14, the specimen being prepared in a similar manner to the aluminum bronze.

STATIONARY BLADING.

Inlet Nozzles.—Commencing first with the inlet nozzles, these are of cast iron on all the turbines dealt with. Fig. 15 shows the upper section of inlet nozzles for one of the 9,600-kw. Rosherville sets. This section is bolted to the face of the nozzle box and registers with eight ports of circular section leading from the latter. The lower section, containing six ports, is bolted to the nozzle box immediately below the upper section. In the nozzle casting the port section gradually changes, and becomes rectangular where the steam leaves the nozzle and impinges on the moving blades. The separate nozzles or ports are divided from each other by a thin wall of cast iron, varying from 3 mm. in thickness on the discharge side, and increasing in thickness as the nozzle assumes the circular section at the inlet side. Fig. 16 shows one of these nozzle sections cut down the center.

As supplied by the makers these nozzles are not at all satisfactory, the dividing pieces or vanes crack badly, and frequently large pieces break out and pass through the turbine; this sometimes occurs after a few thousand running hours, and has been the cause of much anxiety and frequent examination. Usually, beyond slight bruising of the blades, little damage is done, but there is always the possibility of serious injury to the blading, and it is highly probable this was the cause of a partial strip on one of the sets. It is not an easy matter to locate the cause of

many blading troubles, as a piece of metal will frequently pass through several stages and leave very little trace before becoming finally jammed about the center or towards the exhaust end of the machine.

Fig. 17 shows the appearance of a set of nozzles from one of the Rosherville 9,600-kw. sets after being in use for 3,000 hours. Every vane is cracked or broken, but if pieces were knocked out of the two upper vanes, which are apparently sound, and loose portions chipped clear, the nozzles which would be perfectly safe for running purposes. The two vertical marks seen in the whole vanes are not cracks, but saw cuts, and were put in to prevent the damage referred to, but very little improvement, if any, was effected. The photo is hardly representative of the condition of inlet nozzles after a running period of 3,000 hours; usually two or three vanes only in each section have pieces broken off after that running period.

One would naturally think that with a nozzle damaged in this way a considerable falling off in efficiency would result; this, however, was not noticed, and led to a test being carried out with a particularly bad set of nozzles. The result of this test showed that the turbine was as efficient as if nozzles in good condition had been used; it was obviously sound practice, therefore, to remove that part which invariably broke away. A V-shaped portion was accordingly cut out before fitting the nozzle to the machine. Fig. 18 shows the idea, which has been carried out on all the main turbines, since when we have been practically free from breakage of nozzle vanes.

A good deal of bulging and distortion still takes place, due to the expansion and permanent growth of the cast-iron; cracks also develop frequently at the apex of the V, but the vane adheres to the nozzle checks and usually does not break off, as was formerly the case. Cast iron is far from being the best material for inlet nozzles, but it is no doubt very much cheaper in the first place than anything else, and for this reason should receive some consideration before being discarded. A further experiment has been tried, namely, steel blades, forged from mild steel bar to the same shape as the cast-iron vane and cast into an iron body. The analysis of the steel is not known, but the bar is of ordinary commercial quality. This nozzle was made locally, and has been running on one of our large (9,600-kw., Rosherville) turbines for over 3,000 hours. When last examined it was in excellent condition—in fact as good as the day it was first put into commission, and should further use fail to disclose defects in this type of construction, the idea is well worth adopting. There are no thin pieces of cast iron to break off, and although a V-shaped portion is cut out of the vanes in the cast-iron nozzles previously described, and additional safety is thereby attained, there is still the chance that a piece of the remaining portion of the vane may break away and do a lot of damage. There are two exceptions to the general rule of cast-iron nozzles on the plant installed at the power stations—one on a small compressor turbine built by Pokorny and Wittekind, in which the nozzles are neatly carved out of a steel block, and the other on a condensing-set turbine built by W. H. Allen and Co., the nozzle in this case being of steel, machined and inserted in a cast-steel body. These two turbines are not dealt with in this paper. In both cases the result leaves nothing to be desired, and this metal is probably as suitable as any at present in use by turbine builders. Bronze castings have been used by some manufacturers,* but the author is not aware of the composition. Brass would probably be too soft. Monel metal might prove suitable if it would stand the high temperature and give a good, clean casting. Steel castings would be difficult to make

* "Design and Construction of Steam Turbines," Martin, p. 231.

on account of the thin dividing pieces, and it is doubtful whether a sound, smooth casting could be produced.

Curtis Stationary Segments.—Four metals have been used for the blading of Curtis stationary segments, and details of the results when fitted with the various classes of turbines are given below.

1. *Aluminum Bronze* (*L*).—This metal was originally used on turbines Classes 1 and 6, and proved totally unsuitable; the metal disintegrates rapidly owing to temperature, in a similar manner to the wheel blading previously described.

2. *25 per cent. Nickel Steel* (*K*).—This metal has also been used on turbines Class 1, and stands up well for a time, but becomes brittle with use. One failure has occurred, the blade breaking at the narrow part of the neck where fixed to the segment. In the case referred to failure occurred after 33,000 hours. K steel has also been used for the first row of blades on the stationary segments of the auxiliary turbines, Class 6, and one case is on record where the blades failed by breaking at the root after 15,000 hours.

3. *5 per cent. Nickel Steel* (*P*).—This material is now used exclusively for all main turbine stationary segment blading, and stands up very well. In some cases renewal has been necessary owing to wear, the inlet edge particularly becoming concave, and both edges very thin and easily broken. Where more than one row of stationary blades is used, as on turbines Class 2, it is the first row which suffers, but this row does not require reblading under 20,000 hours. The second row has not, so far, been rebladed, the maximum running hours being 30,678. In the auxiliary turbines, Classes 6 and 7, P steel is nearly always used in the first row, and frequently in the second.

4. *Brass* (*M*).—Brass is sometimes used for the second row of the auxiliary turbines, Classes 6 and 7, and, under present practice, always for the third row. It stands well, the maximum hours in use so far being 30,673.

It will be seen from the above that the metals used for Curtis stationary blading behave in the same manner as when used for Curtis wheel blading.

Guide Blading.—In introducing the subject of deterioration of these blades it will be as well to commence with a description of what actually happens to the blade, and afterwards take a few typical cases. Thirty per cent. to 32 per cent. nickel steel (J) is the material in general use, and, unless otherwise stated, the deterioration refers to this material. The nature of the deterioration is the same in all cases, but the results, taken over a given period, are quite inconsistent, and vary considerably with different turbines in the same station, and show a still greater difference with similar machines at different stations. The first visible indication appears in the form of a fine crack, always on the inlet side of the blade, and frequently close to the outer retaining ring, but sometimes near the center of the blade. Before these cracks are actually visible they can be produced by a few light hammer taps. These cracks extend over the surface of the blade, cobweb fashion, until pieces become detached and pass through the succeeding stages, severely bruising and closing up the wheel blading on the inlet side, but seldom causing a strip; the amount of damage done is naturally regulated by the size of the broken piece. All blades in the guide ring are not affected at once; generally, isolated blades distributed round the ring develop these cracks, but often the defective blades appear in batches; ultimately the whole of the blades crack, and, if not attended to, the guide ring collapses, causing heavy fouling and serious damage to the wheels and blading.

Fig. 19 shows a portion of a guide-blade diaphragm, the cracks in the blades being clearly visible; this had been in commission for 9,391 hours

TABLE VI.—GUIDE BLADE RINGS FOR 9,600-KW. TURBO-GENERATORS, CLASS 2. RUNNING HOURS T
OR TO TIME OF RENEWAL.

All blades 3 mm. thick. 148 blades per complete ring for all stages.
All blades are of 30 per cent. to 32 per cent. nickel steel. J except where marked H. S.R. denotes still running. X signifies re

		Stage.										
Length of Blade ..		2. 19 mm.	3. 24 mm.	4. 30 mm.	5. 36 mm.	6. 47 mm.	7. 63 mm.	8. 87 mm.	9. 119 mm.	10. 160 mm.	11. 199 mm.	1 250
No. 1, Rotherville	Original ..	23,937 S.R.	23,937 S.R.	23,937 S.R.	23,937 S.R.	15,230 X 8,707 S.R.	9,900 X 14,037 S.R.	15,230 X 8,707† S.R.	15,230 X 8,707 S.R.	15,230 X 8,707 S.R.	20,909 X 3,428 S.R.	9, 14, S.
	1st renewal	—	—	—	—							
No. 2, Rotherville	Original ..	29,909 S.R.	29,909 S.R.	29,909 S.R.	29,909 S.R.	19,887 X	15,908 X	19,887 X	19,887 X	19,887 X	19,887 X	19,9 X
	1st renewal	—	—	—	—	9,722 S.R.	9,722 S.R.	9,722‡ S.R.	(In state of collapse when renewed.) 9,722 S.R.		9,722 S.R.	9, S.
No. 3, Rotherville	Original ..	20,862 S.R.	27,242 S.R.	27,242 S.R.	27,242 S.R.	27,242§ X	18,023 X	18,023 X	24,403 X	24,403 X	24,403 X	27, X
	1st renewal	—	—	—	—	—	9,219	9,219	2,839	2,839	2,839	—
							S.R.	S.R.	M.S. Rebladed rings-blades. S.R.		Flanged	
No. 4, Rotherville	Original ..	2,041 Recently fitted H —	31,926 S.R.	31,926 S.R.	31,926 S.R.	12,075	12,075	12,075	12,075	12,075	12,075	12,
	1st renewal					19,851 H S.R.	Breakdown of machine 19,851‖ S.R.	19,851 H S.R.	damaged blades stages. 19,851 H S.R.		Cause unascert 19,851 H S.R.	19, S.
No. 5, Rotherville	Original ..	4,670 recently fitted	20,275 mech. damage	24,945 S.R.	20,275 mech. damage	20,275 mech. damage	top 12,474 X bottom 17,110** X	top 12,474 X bottom 17,110** X	17,110	17,110	17,110	20,
	1st renewal	—	4,670 S.R.	—	4,670 S.R.	4,670 S.R.	top 7,501†† bottom 3,165	top 7,501†† bottom 3,165	7,335 S.R.	7,335 S.R.	7,335 S.R.	4, S.
	2nd renewal	—	—	—	—	—	4,670 S.R.	4,670 S.R.				
No. 1, Verseniging	Original ..	26,027 S.R.	For all stages. All guide rings ceased after running 15,485 hours. Cracks produced on sample of blade from 11th stage on heavy jarring with chisel after 20,706 hours. Cracks produced in stages 9, 10 and 11 blades on tapping with hammer after 24,000 hours.									
No. 2, Verseniging	Original ..	21,899 S.R.	For all stages. Guide rings ceased after 9,467 hours. Cracks produced on stages 10 and 11 blades on tapping after 20,000 hours. Not visible otherwise.									
No. 3, Verseniging	Original ..	2nd stage not fitted	25,564 S.R.	For all stages. Guide rings ceased after 15,174 hours. Cracks produced on stages 7 to 11 blades on tapping after 20,000 hours. Not visible otherwise.								
No. 4, Verseniging	Original ..	2nd stage not fitted	20,503 S.R.	For all stages except 10. Cracks produced on Stage 11 on tapping blades after 16,919 hours. All guide rings ceased after 9,273 hours.						16,919 X 2,589 S.R.		
	1st renewal											

...G HOURS TO SEPTEMBER 30, 1916.

X signifies renewed owing to deterioration.

11. 199 mm.	12. 250 mm.	13. 290 mm.	Remarks.
20,500 X 2,428 S.R.	9,900* X 14,037 S.R.	20,500 X 3,428 S.R.	*Stage 12 renewed owing to mechanical damage. †One blade cracked when last examined.
19,887	19,887	19,887	
X hen renewed.) 9,722 S.R.	X 9,722 S.R.	X 9,722 S.R.	‡One blade in stage 8 developed cracks after smart tapping.
24,403	27,242	27,242§	
X 2,999 —Flanged	X —	X —	§Renewed recently—not yet recommissioned.
12,075	12,075	12,075	
Cause un 19,851 H S.R.	ascertained 19,851‖ S.R.	19,851 H S.R.	‖Cracks produced on blades in stages 7 and 12 when hammered smartly.
17,110 7,235 S.R.	20,275 4,670 S.R.	24,945 —	**The bottom halves of guide rings for stages 7 and 8 were not renewed at the same time as top halves. ††Stages 7 and 8 damaged mechanically.

only, on one of the compressor turbines, Class 5, and failed a few months after this photograph was taken owing to lateral movement of the diaphragm having taken place in the direction of the following wheel, which was ultimately fouled.

Fig. 20 shows a portion of the upper half of the guide ring for the eighth stage removed from No. 5 generator turbine, Rosherville, after 7,801 hours. Reference to Table VI, giving the life of guide blades for Class 2 turbines, shows this to be due to mechanical damage caused by a partial strip; the mechanical damage is, however, slight, and an examination of the ring is interesting, nearly every blade being fractured where joining the inner and outer retaining rings. The photo is not clear, but the light splashes near the lower ring show where fracture has taken place, and, in addition, several blades have holes right through them, more particularly at the junction between the curved portion and the straight. The portion of blade clear of the retaining rings looks quite good until smartly tapped with a hammer, when the characteristic cracks showing early deterioration are produced. The lower half of this ring, which had only run 3,165 hours, was removed and carefully examined, but it was found to be quite sound.

Fig. 21 shows three loose guide blades from generator turbines, Rosherville, in various stages of deterioration. (*a*) (Specimen 2/15) was removed from seventh stage guide ring No. 2 set after 15,308 hours; (*b*) (specimen 4/15) from seventh stage No. 3 set after 18,023 hours; (*c*) (specimen 12/15) from twelfth stage No. 2 set after 19,887 hours. These blades are in the same condition as when in position in the guide ring, and have not been further damaged during removal. Fortunately, in the case of the guide-blade rings under discussion, there is little danger when the cracks first appear, as several months elapse before pieces of guide blade break away. Examination at intervals of 3,000 hours allows one to follow the condition and judge when the ring should be renewed.

In the case of the guide-blade diaphragms, as fitted to Class 4 and 5 turbines, they become unsafe as soon as cracks appear, and movement of the diaphragms in the direction of the following wheel has taken place on different occasions This tendency to displacement of the diaphragm is probably due to the circumferential pitch of the blades in the guide-blade diaphragm construction being greater than in the guide-ring construction, with a consequent increase of diaphragm area per blade. The subjoined table gives the diaphragm area exposed to the difference of steam pressure between the inlet and discharge side, also the number of blades and pitch for the first set of guide blading, *i.e.*, for stage 2. Three classes of turbines are dealt with, the diameter being taken to the nearest inch. There are 12, 11 and 4 Rateau stages, in Classes 2, 4 and 5 turbines respectively, and assuming the pressure drop across each stage to be 6 pounds per square inch for the first two, and 12 pounds per square inch for the third, the weight carried per blade is shown in the right-hand column.

	Area of diaphragm.	No. of blades.	Pitch of blades.	Weight carried per blade.
	sq. in.		mm.	lbs.
9,600-kw. turbines, Class 2.....	4,624	148	42.0	187
12,000-kw. turbines, Class 4....	3,606	76	74.6	284
10,000-h.p. compressor turbines, Class 5.....	2,109	52	84.0	486

Fig. 22 shows the difference in the shape of the blades for a guide-ring and a guide-blade diaphragm. It will be noticed that the curved portion marked X is much longer in the former than in the latter construction.

With regard to the life of these blades, Table No. VI gives the number of running hours up to the time of replacement, if such has taken place, or up to September 30, 1916, if the original rings are still in use. All the turbines are of 9,600 kw. (Class 2), five being in use at Rosherville and four at Vereeniging. The conditions are generally the same at each station, with the exception of the steam temperature, which is 350 degrees C. at Vereeniging, as against 320 degrees C. at Rosherville. Reference to the table shows the great disparity in the running hours between the various stages at the same station, and the same stages at the two different stations.

First, it will be noticed that in no case has it been necessary to renew the first four guide rings owing to deterioration; in one case renewal has taken place; this was, however, due to mechanical damage owing to fouling some hard material, probably a piece of nozzle vane. It will also be noticed that deterioration commences first in stage 7 on the first five sets, and that the life varies from 9,900 hours on No. 1 to 18,023 on No. 3. As in the case of the wheel blading, the "life" does not represent the time the blades would actually have lasted; it simply shows that when renewed the blades were defective, the degree of deterioration varying considerably. Generally speaking, no rings were renewed until small pieces of blades broke away, and in some cases the rings were in an absolute state of collapse before being renewed.

Allowing for this difference in condition, however, there is no satisfactory reason to account for a difference of over 8,000 hours in the life of stage 7 on sets 1 and 3. The variation in the life of the center stages on the Rosherville and Vereeniging sets is very marked, and before attempting to explain this it will be well to consider possible causes for this deterioration.

Influence of Material on Deterioration.—The evidence is hardly sufficient to state definitely the relative advantages of high or low percentage nickel steel, but the results obtained in No. 4 set, Rosherville, point to steel containing no nickel as being more suitable. Stages 6 to 13 on No. 4 were tested in exactly the same manner, *i.e.*, they were struck sufficiently hard with the ball of a hammer to indent the blade in several places where exposed at the horizontal joint. Cracks could only be produced on stages 7 and 12, which on analysis were found to contain 30 per cent. nickel; the other stages resisted this somewhat drastic treatment, and these blades were marked H by the makers and found to contain no nickel. When last examined there was a good deal of pitting on the convex and concave portions of these H blades, which effect is unusual with the nickel steel blades.

We are experimenting with different classes of steel; Siemens-Martin, and 5 per cent. and 3 per cent. nickel steel being at present in use, cast into the retaining rings in a similar manner to that adopted by the makers. The test is of far too short a duration to form an opinion as to the respective merits of these various steels, and the time required is one of the drawbacks to these experiments, as, even when the turbine is run at a very high load factor, two and a half to three years must elapse before we can say that a definite improvement has been made, unless, of course, the steel on trial gives much worse results than that at present in general use.

On the Class 4 turbines H steel, containing no nickel, has been supplied for some stages, and has so far shown no deterioration, while other stages, containing 30 per cent. nickel (J steel) in the same machine have shown marked deterioration, but these will be referred to later.

In the case of the guide ring renewed on No. 4 set, Vereeniging (stage 10), some of the blades were badly cracked, and others appeared to be in perfect condition; one of each was analyzed, and both were found to be 30 per cent. nickel steel, the inference being that the presence of a high percentage of nickel does not necessarily cause the blades to crack.

A non-ferrous material might be tried with good results. The cost would be a good deal higher than if steel were used, but that would not matter much if the life were considerably increased. Casting iron round blades made from Monel metal sheet has been tried, but the heat of the molten metal appears to take the nature out of the copper-nickel alloy, rendering it short, so that it cracks and breaks on bending.

Mechanical Stresses imposed on the Blades by Expansion and Permanent Growth of the Retaining Rings and Diaphragms and by Sudden Changes of Temperature in the Blades.—We have ample evidence of the exceedingly heavy stress imposed on the blades from this cause, particularly at the high-temperature end of the turbine.

When the machines were first erected the guide rings were fitted to a groove in the casing, and secured thereto by means of screws; the C.I. diaphragm, of dished formation, is grooved round the periphery and fitted snug to the inner retaining ring of the guide ring, there being a corresponding tongue fitting the groove of the diaphragm to prevent lateral movement. The convex side of the diaphragm is toward the high-pressure end of the machine, and is designed to resist the difference of pressure between the inlet and discharge side of the guide rings. It may have been assumed by the makers that this dishing would relieve the blades of excessive stress by deformation of the casting, and the latter effect actually occurs, as it is found that the clearance between the diaphragm boss and the preceding wheel gradually diminishes. It, however, does not relieve the blades sufficiently to prevent severe buckling of the first two rings, and Fig. 23 shows this very clearly. Photographs of the 9,600-kw. machines are not available, but the figure shows a portion of a stage 2 ring removed from one of the Simmer Pan 3,000-kw. sets, the deformation of the blades in the larger sets being similar, although not usually quite so pronounced.

In addition to the buckling of the blades, they are badly cracked, as seen in Fig. 24, due doubtless to the enormous stress imposed. This particular ring was in commission for 39,000 hours, and is the first one renewed on the early stages of the machine owing to cracked blades. On the 9,600-kw. sets these cracks have not yet appeared on the first five stages.

It is seldom the blades are buckled beyond the third guide ring, the expansion of the guide rings and diaphragms being naturally less as the temperature decreases, until at the exhaust end it is comparatively low, seldom exceeding 50 deg. C. If we assume heavy pressure on the blades to be the cause of the cracking, it seems strange that where the pressure is heaviest the blades are least affected. It may be a contributory cause, but in any case it is obviously better to relieve the blades of this stress as far as possible, and this has been accomplished in the case of the guide rings by machining a 3 mm. clearance between the guide ring and diaphragm and resting the diaphragm on two short fitting strips on the lower half. A few copper pins are fitted on the upper half of the guide ring, with about 0.5 mm. clearance between the pins and diaphragms, so that the latter cannot rise too much and foul the shaft gland. It is found that after a time these pins are crushed by the diaphragm, and they are eased periodically.

In the case of the guide blade diaphragms these are suspended on lugs at each side of the turbine casing, 3 mm. clearance being left all round the casing groove, so that the diaphragm is free to move when

fitted, but gradually tightens sideways in the groove as the growth of metal takes place.

Reference to the notes in remarks column of Table VI shows that in sets 1 and 2, Rosherville, cracked blades are found in the eighth stage after 8,700 and 9,700 hours. Also in the eighth stage of No. 5 set, Rosherville, after 7,801 hours. These three rings were eased when first fitted and have never been subjected to heavy pressure. It therefore seems clear that the metal deteriorates irrespective of the compression to which the blades are subjected.

In the case of the guide-blade diaphragms on turbines Classes 4 and 5, which were fitted to relieve blade stresses, 1 mm. clearance was left round the circumference by the erectors. This was absorbed during the first year of running, and, in addition, in some cases, very heavy distortion has taken place.

Fig. 25 shows the guide-blade diaphragm for the second stage on one of the 12,000-kw. sets installed at Brakpan in 1915, and the photograph was taken after 8,822 running hours (14 months). The rim of the diaphragm portion should be flush with the top side of the outer retaining ring, but has moved 3/16 inch toward the following stage, shown by the dotted line. In addition, practically every blade is cracked, this being the first instance which has occurred at such an early stage in the machine after such a short life; the material is 30 per cent. nickel steel, 2 mm. thick. In this case the clearance was most likely very soon taken up, and the heavy pressure doubtless had much to do with the cracking and was mainly responsible for the distortion.

Finally, we have cases on record where long life has been attained without the rings being eased at all. On one of the 3,000-kw. sets installed at Brakpan some of the blading was renewed after 46,890 hours (stages 2, 6, 7), and on the adjoining set the blading was eased after over 40,000 hours, and is still in commission and in good condition after 59,000 hours, with the exception of the guide ring for stage 2, the blades of which are now cracking. In both machines the blades are of J steel. The steam temperature at the stop valve has been somewhat lower on these sets, being about 290 degrees C., as against 320 degrees C. to 350 degrees C. on the sets previously dealt with. This difference in life is most difficult to account for. Some makers use steel castings for turbine diaphragms, and this material possesses an advantage over cast iron in that it is not, as far as the author is aware, subject to permanent growth to the same extent as the latter material.

With regard to the sudden expansion of the blades themselves when starting up a set from a cold state, the blades are thin and will temporarily attain a higher temperature than the retaining rings; they will be subjected to compression stresses on this account, although 30 per cent. nickel steel probably has a very low coefficient of expansion; 36 per cent. nickel steel (invar), used for instrument work, has no coefficient of expansion.

In this connection the number of starts over an extended period have been examined on the various 9,600 kw. sets, but this does not seem to have any bearing on the life of the blades, and it is probable that these stresses are not so serious as others which the blades are called upon to withstand.

It having been decided to cast these guide blade rings locally, their manufacture presented some difficulty at first, and previous to the work being put in hand some of the old guide rings were rebladed as an emergency measure. Ordinary steel plate, 2 and 3 mm. thick, was used, cut to shape and bent round a former. The blades were then flanged and riveted between the retaining rings, a fairly neat job being made. Fig. 26 shows the blade ready for fixing. The blades were quite sound

when last examined, but had only been in commission for 2,839 hours. It will be interesting to see whether this method of building up the rings will increase the life of the blades. There may be some advantage in the flanging, a little more elasticity, perhaps, and, further, the blades are not subjected to the intense heat which is unavoidable when the molten iron is cast round them. It is significant that the turbine in which these rebladed rings are fitted is less efficient than the others of similar type. This may be due to the rivet heads and flanged portion of the blades, which project into the steam passage and obstruct the steam flow.

Other Stresses, Vibration, Impurities in the Steam, &c.—If we assume the drop in pressure across each diaphragm to be 6 pounds per square inch, and the diameter of the diaphragm to be 60 inches, the guide blades have to withstand a lateral force of 8.4 tons, which, divided between the blades, may not be excessive. In addition, however well the machine is balanced, a certain amount of vibration will take place, due to the rotation of the parts, and also due to the passage of steam through the blading. This vibration, acting on the guide blades while under stress from the diaphragm pressure, and perhaps at a critical temperature, may cause cracks to develop and ultimately break down the material. Another factor which may influence deterioration and account for the difference in life at various stations is impurities carried over from the boilers with the steam. For instance, on the 3,000-kw. sets at Brakpan, where exceptionally long life has been attained, the boilers are supplying steam to reciprocating compressors, and oil used for lubricating the steam cylinders has found its way into the boilers, and from these to the turbines. It occurs here as a greasy deposit on the guide blades, which may have acted as a protective covering. The feed-water make-up at this station was at one time strongly alkaline, being quite different from that at Rosherville or Simmer Pan, while at Vereeniging river water is used, which is comparatively soft.

Two final cases are given, which are interesting on account of the rapidity with which deterioration takes place, and also because the machines are of later design than those previously described. The first of these is one of the 12,000-kw. generator turbines installed at Brakpan, containing 11 guide blade diaphragms, the blades in the first seven being 2 mm. thick, of high percentage nickel steel J; of these the 1st, 6th and 7th contained blades so bladly cracked after 8,822 hours as to be unsafe, while in the 5th the blades were just commencing to crack; in diaphragms 2, 3 and 4 the blades were quite sound; in the remaining four stages, beyond the 7th stage, the blades are of steel containing no nickel (H) and are in good condition; in three of these the blades are 3 mm. thick, and in the fourth and last 4 mm. thick. The first five stages on this set have been subjected to heavy pressure, so much so that displacement of the diaphragms has occurred, as shown in Fig. 25, previously described. It would appear from the above that the H steel, 3 mm. thick, will give better results than the 2 mm. J steel, although it is strange that in diaphragms 2, 3 and 4 the latter material shows no cracks; one can only conclude that there must be some difference in manufacture of the plates from which the blades are made, as the rough analysis shows the material to be similar, and the conditions are the same as on the diaphragms at either side.

The other case is on the three large compressor turbines of 10,000 H.P. installed at Rosherville.

There are five guide blade diaphragms in these sets, the first two containing H steel blades without nickel, 5 mm. thick on the inlet side, tapering to 2 mm. on the outlet side. These blades show no cracks after 11,000 hours. Diaphragms 3, 4 and 5 contain blades 2¼ mm. thick, of

uniform section, and on two sets diaphragm 3 has nickel steel (J) blades, which have both cracked badly after 10,000 and 11,000 hours respectively. On the third set blades containing no nickel (H) are used, and at present are quite sound after 6,000 hours. On the fourth diaphragm J steel has been used for compressors 8 and 9, and in both cases the blades have cracked after 11,000 and 6,000 hours respectively; while on compressor 7 H steel is used and is in good condition after 10,000 hours. J steel is used throughout for diaphragm 5, and when examined the blades were cracked in one case only.

The following table shows the arrangement of blades and condition when examined after the number of running hours specified. H signifies steel containing no nickel, and J 30 per cent. nickel steel:

10,000-H.P. COMPRESSOR TURBINES.

Diaphragm No.	No. 7. 10,362 hours.	No. 8. 11,543 hours.	No. 9. 5,994 hours.
1	H, steel good	H, steel good	H, steel good.
2	H, steel good	H, steel good	H, steel good.
3	J, cracked	J, badly cracked	H, good.
4	H, good	J, badly cracked	J, slightly cracked
5	J, good	J, cracked	H, good.

The above results point to H steel being a much more suitable metal for guide blading than J steel. No. 7 compressor has been opened up quite recently (since the above table was drawn up), and the guide blades have been hammer-tested severely. In the third diaphragm cracks in the blades are clearly visible before hammering, and with a few smart blows the cracks extend over the surface and pieces can be knocked out. The H steel blades, containing no nickel, used in diaphragm 4, were treated in the same manner, but no cracks were produced, the metal being quite sound. In diaphragm 5 no cracks were visible until the blades were struck a few times, when the cracks appeared, showing the metal to be in the early stages of deterioration. The guide blading was in commission for just over 13,000 hours when this examination took place.

A photomicrograph of the H steel in use is shown in Fig. 14.

Conclusions.—In summarizing the behavior of the materials used in blading these turbines, it will be seen that there is no difficulty in choosing a material for moving blades which can be depended on to give fairly satisfactory results in particular stages. Five per cent. nickel steel, brass and aluminum bronze in suitable stages will have a life of 40,000 hours, and perhaps 50,000, or more, which, if the turbines are run continuously, mean five or six years. It is probable that in the majority of power stations the running hours would be much less, and the number of years the blades would last will be correspondingly increased. One has to decide what constitutes a satisfactory life, and if we assume the turbine will be in use for 15 or 20 years, the ideal blade material is one which will do its work efficiently for this period. If long life can only be achieved by the use of expensive alloys, the cost of such material would have to be compared with the saving in maintenance which would result by its use.

With regard to the life of turbine plant one has to consider the probability of future improvements, which might increase the efficiency

of turbines to such an extent that it would pay to scrap existing plants and install new ones.

One has to admit, however, that a very considerable advance in efficiency would have to be made to justify this course, and the uncertainty of a sufficient improvement is a further argument for better blading material.

A saving of 1 per cent. in coal consumption on a 10,000-kw. set supplying 6,000,000 units per month may represent about 400*l.* per annum.

With regard to inlet nozzles, it is clear that no reasonable expense should be spared in making these absolutely reliable; nozzles with thin cast-iron vanes should be abolished, but if cast iron is used the design should be such that there is no possibility of pieces becoming detached.

The guide blading in use on these turbines is clearly unsatisfactory, and the reader may wonder why this state of affairs has not been remedied long ago. The user is, however, largely in the hands of the turbine builder, and, further, if a mistake has been made in design or material in the original installation, three years or more must often elapse before this is discovered, and an equivalent time is taken to prove the success or otherwise of any alteration. Incidentally, thousands of pounds are spent by the user in large installations in renewing defective blading and making good damage caused by broken pieces, collapsed rings, &c. This maintenance should be unnecessary, and for this reason the defects have been given in considerable detail.

While the author feels that mild steel is preferable to the high-percentage nickel steel in use, it must be admitted that the latter has given good results in many cases. On the fourth diaphragm J steel has been used for the 10,000-H.P. compressors 8 and 9, and in both cases the blades cracked, while on compressor 7 H steel is in good condition.

High-percentage nickel steel has been more extensively used and for a much longer period than non-nickel steel. The latter material is in use on No. 4 generator turbine, Rosherville, and in this case considerable pitting has occurred. The evidence obtained on the 10,000 H.P. compressor turbine, Rosherville (No. 7), is, however, very much in favor of mild steel, and there seems little doubt that it will prove the better material. The possible causes of deterioration are not conclusive; they may all be contributory to the general failure, and the inconsistency of results, particularly at different stations, makes the whole subject very puzzling. One is inclined to blame 30 per cent. nickel steel because so much of it is in use and fails, and it may be that mild steel will also prove unsatisfactory through cracking or pitting.

Blades made from brass sheet and carefully fitted to retaining rings have been used for small turbines by some manufacturers, and various methods of attaching blades could be devised. The cast-in method used on the turbines described in this paper is, however, very simple and requires a minimum amount of machining and fitting, so that it would be preferable to retain this design if satisfactory results can be obtained.

The author believes that the position can be much improved by the use of thicker blades of mild or low-percentage nickel steel, say, 3/16 inch, or ¼ inch tapered, perhaps, on the outlet side to ⅛ inch. Unfortunately an alteration of this kind on a ring designed for 2 mm. blades reduces the area for steam passage somewhat. To obtain the same port area the radial length of the guide blade would have to be greater, necessitating a longer moving blade. The following alterations have been made, which may result in a considerable improvement in the life of the guide blades.

In the case of the guide-blade diaphragms for the 10,000-H.P. compressor turbines, in addition to substituting B, C and C1 steel for J steel, the blades have been increased in thickness from 2.25 mm. to 3 mm. in three diaphragms on one turbine without affecting the efficiency or output.

On the 12,000-kw. generator turbines at Simmer Pan and Brakpan six stiffening blades of mild steel, 3/16 inch thick, have been cast in each half; these are distributed equally round the diaphragms and take the place of the J steel blades, 2 mm. thick, originally used. The remaining blades are 2 mm. thick, B, C and C1 steels being used in the various diaphragms.

On the same class of turbine another experiment is in hand; cast-iron stiffening blocks are being cast between adjacent guide blades on the horizontal and vertical diameters of the diaphragm. This blocks up four ports in the complete ring, and should stiffen up the diaphragm very considerably. In addition, the material has been changed from high-percentage nickel steel to B and C steel.

The guide-blade diaphragms on which the last two experiments are being tried are not yet in commission, so that the effect on efficiency is at present unknown.

The author cannot say whether similar trouble is prevalent on turbines made by other firms, and any discussion which this paper may promote will be valuable in throwing light on the best design and material for turbine blading. It is hardly to be expected that manufacturers will advertise defects in their machines, but users need have no scruples in this respect.

In conclusion the author desires to thank the management of the Victoria Falls and Transvaal Power Company, Limited, for permission to publish the information contained in this paper, and also Messrs. C. W. R. Campbell and Val Davies for preparing the slides and photographs which accompany the paper.—"Engineering."

"THE DETERIORATION OF TURBINE BLADING."

To the Editor of "Engineering."

Sir:—The interesting and instructive paper on the above subject recently read before the South African Institute of Engineers by Mr. A. Fenwick, and published in your issues of May 4 and 11, has justly received a wide publicity in the technical press of this country, and it is to be hoped that our manufacturers and users of turbines will take advantage of such a complete and thorough statement of working results. There is, however, a danger that, in the case of those who are not thoroughly familiar with all the circumstances, doubts and fears as to the reliability of impulse turbines may arise, which will tend unjustly to discredit this type of turbine as built in this country. In the hope that these fears may be removed, we trust that we, as the owners of the Curtis turbine patents for land purposes in this country, may be allowed to place before your readers a few words of explanation and information.

In the first place, we are pleased to note that in the preamble to your reprint of Mr. Fenwick's paper reference has already been made to the fact that all the turbines are of one particular firm's make, and that the failures recorded are not due to the type of turbine, but to the use of improper materials or design.

When we took up the manufacture of Curtis turbines about fourteen years ago the blades were cut from solid rings—the material was mild steel, and was found to be serviceable, particularly in the high-pressure stages. Towards the exhaust end, owing to an increasing amount of moisture in the steam, erosion would commence after some 6,000 hours' service. This deterioration was augmented in some cases by ordinary rusting taking place while the turbine was out of service, and due to

leaky stop valves, the blading standing in a warm damp atmosphere. Many of these early turbines are still running, and have suffered less in efficiency than would be expected from the appearance of the blading, thus confirming Mr. Fenwick's observations. It was early recognized that a more suitable material was wanted for the blades, and systematic investigations were started by us to find a material possessing the following characteristics:

1. Non-corrosive in steam and water.
2. Resisting erosive action of high steam velocities.
3. High tensile strength.
4. Retention of initial physical properties.
5. Adaptability for manufacture.
6. Reasonable cost.

It will not be necessary here to relate the methods employed, or give all the results of our investigations. Those interested may be referred to a paper read by our chief chemist, W. B. Parker, F. I. C., before the Institute of Metals ("Journal of the Institute of Metals," 1915, vol. xiv), written for the purpose of soliciting the active assistance of the metallurgists towards getting the most suitable material for turbine blading.

Among the non-ferrous alloys we have found that high-grade phosphor-bronze best filled the above-given requirements, and after several years' experience we have found none better. This alloy has initially a yield-point of 12 tons and tensile strength of 24 tons. After being exposed to high steam temperature the alloy ages, *i. e.*, the yield-point is lowered, and somewhat in proportion to the temperature to which it is exposed. This reduction in strength is, fortunately, not important, as the shortest and least stressed blades are exposed to the greatest heat. We believe the phosphor-bronze has lately been adopted for impulse marine turbines, and with satisfactory results.

The output of steam turbines for a given r.p.m. is gradually growing, and the blading stresses have correspondingly increased, and to such values that the phosphor-bronze alloy is not strong enough for the requirements at the low-pressure end of the turbine. Where this greater strength is required, steel or steel alloys are used.

To avoid corrosion high-percentage nickel steel looked most promising, and was experimentally tried by most turbine builders. Two machines which we built were provided with 25 per cent. nickel steel blading, which came to grief after less than a year's service. The behavior of this alloy is most mysterious, as pointed out by Mr. Fenwick, and we believe there is no satisfactory explanation today as to the real causes of failure, our own investigations being inconclusive. Even with so low a percentage of nickel as 3 per cent. to 5 per cent., trouble may be experienced owing to the tendency of nickel to segregate when pouring the ingot, and we have therefore for some time past been using a high-grade open-hearth carbon steel (having a higher percentage of manganese than usual), so as to obtain a material of great tensile strength, dependable and giving constant results. This steel is not affected by corrosion to any greater extent than steel containing a permissible amount of nickel, and we find that corrosion is not appreciable if the feed water is pure and no deleterious compounds are introduced into the boiler, and, further, that the turbine is kept dry while standing.

It would be desirable, however, if a non-rusting blading material, having great tensile strength, could be obtained. Experiments are in progress in this direction with what is known in the cutlery trade as "rustless steel," but it is too early yet to say if this alloy will stand the test of time.

With regard to nozzle and guide-blade material, there is no doubt that the serious troubles described by Mr. Fenwick are mostly due to employment of 30 per cent. nickel steel alloy, and partly to too thin guide blades.

First-stage nozzles exposed to high pressure and temperature with cast-iron division walls, as mentioned by Mr. Fenwick, could only be expected to give the trouble experienced. Bronze alloys have been found most satisfactory for this purpose, especially when used for extra high temperatures, and the extra cost involved is well justified. The most suitable material for guide blades, according to our own experience, is a good grade of mild steel. If a non-rusting steel can be obtained for the longest rotating blades, this material will be equally useful and welcome for the guide blades.

The drop in pressure across a diaphragm is considerable in the first stages of a turbine. Mr. Fenwick mentions a drop of 6 pounds across the diaphragms of turbines in Classes 2 and 4. It should be pointed out that this is the average pressure-difference, and in the case of the first diaphragm the drop is about four times this amount, increasing the stress per blade to four times that given in Mr. Fenwick's table. We believe that cast-steel diaphragms should be used where the pressure drop is great, the diaphragm being provided with suitable portholes, and having separate nozzle segments bolted to the diaphragm. With this construction all serious bending stresses in the guide blades are avoided, and the extra expense, we think, is justified. Where cast-iron diaphragms are used the metal should be carefully selected so as to avoid excessive growth, which is mostly due to the presence of too much silicon.

In conclusion we would express the hope that the proposed national investigation of blading material will be taken in hand without delay.

(For the British Thomson-Houston Co., Ltd.),
FREDERICK SAMUELSON.

Rugby, May 15, 1917.

SOME EXPERIMENTS ON THE INFLUENCE OF RUNNING BALANCE OF PROPELLERS ON THE VIBRATION OF SHIPS.

By J. J. KING-SALTER, R.C.N.C.

(*Institution of Naval Architects; March*, 1917.)

The experiments to which reference will be made were carried out in two destroyers and a "Town" class cruiser which were being built at the Commonwealth Naval Dockyard, Sydney.

The first destroyer to be completed ran her trials very satisfactorily, but there was a certain amount of small vibration in the vessel, most pronounced right aft above the propellers. That this vibration was due to the main engines was considered most improbable, as these latter apparently ran as smoothly and as free from vibration as they did in the shop test. The only other rotating masses that could possibly be supposed to cause vibration were the propellers. The question then arose, were the propellers, as manufactured, fulfilling the conditions for perfect running balance which, quoting from Mr. Barnaby's book on "Marine Propellers," are given to be:

(1) Each blade must be of the same weight.

(2) The center of gravity of each blade must be at equal distance from the axis of the propeller.

(3) The center of gravity of each blade must lie in a plane at right angles to the axis.

There is no way known to the author for conditions (1) and (2) to be obtained except by careful and accurate workmanship, when condition

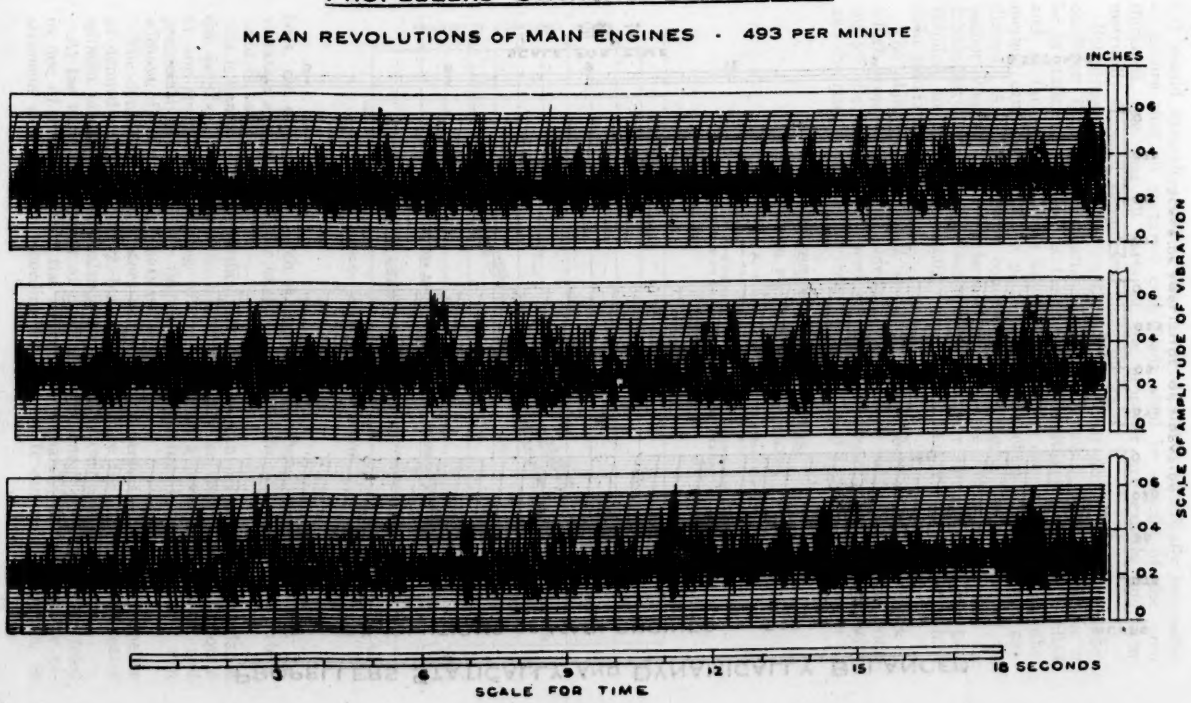

PROPELLERS STATICALLY BALANCED.
MEAN REVOLUTIONS OF MAIN ENGINES · 493 PER MINUTE
INCHES
06
04
02
0
SCALE OF AMPLITUDE OF VIBRATION
3
6
9
12
15
18 SECONDS
SCALE FOR TIME

Fig. 1.

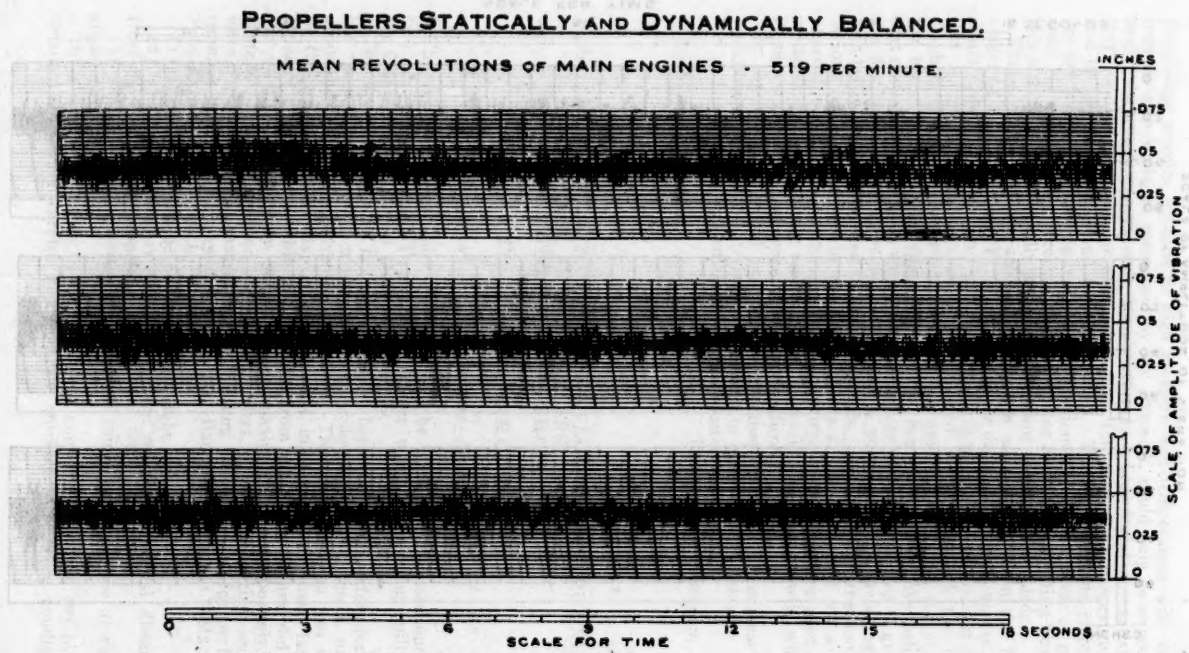

PROPELLERS STATICALLY AND DYNAMICALLY BALANCED.
MEAN REVOLUTIONS OF MAIN ENGINES - 519 PER MINUTE.
INCHES
.075
.05
.025
0
SCALE OF AMPLITUDE OF VIBRATION
0
3
6
9
12
15
18 SECONDS
SCALE FOR TIME

Fig. 2.

(3) would naturally follow. But in practice it is practically impossible to ensure that all parts of each blade shall be exactly of the same thickness, and that all parts of the propeller are exactly and symmetrically disposed around and along the axis.

Balancing the propeller on knife edges does not ensure that either condition (1) or (2) is satisfied, as it is possible for the propeller to be in static balance and yet for these two conditions not to be fulfilled, and, so far as the author's knowledge goes, he does not see what other method can be adopted to place an absolute check on these two conditions. And, as regards condition (3), hitherto, also so far as the author knows, no other check other than the static balance and the calibration has been exercised to prove whether this condition has been fulfilled, and it has apparently been assumed that the above checks have been satisfactory.

The object of this paper is to show that there is another check to which propellers can be subjected, viz: dynamic balancing, *i.e.*, the rotating of the propeller at speed on spring bearings, noting the vibration, and the removal of weight from certain parts of the blades and even the

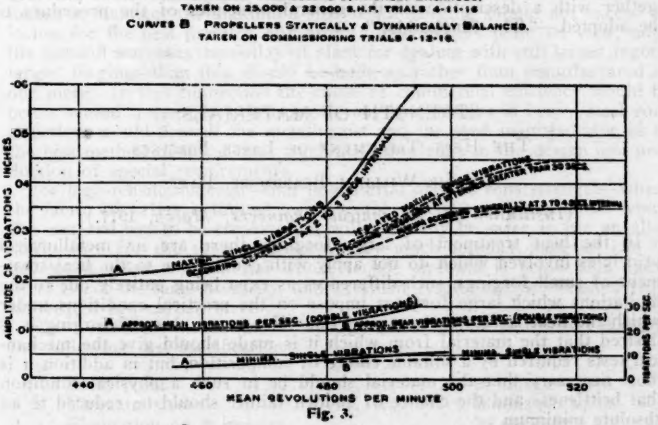


Fig. 3.

boss, the position of which is indicated by the direction in which the vibration acts, in a manner exactly similar to the way in which a turbine rotor is balanced.

It is contended that by this further balance a perfect running balance can be obtained, *i.e.*, one in which the propeller can be made to rotate at any speed in the spring bearings without a perceptible tremor, and presumably, if it will do this in a live bearing, it will similarly run without vibration on its shaft in the ship.

The dynamical balance does not provide a perfectly balanced propeller as defined by Mr. Barnaby, but it does provide for one that will rotate without vibration if the balancing is properly carried out.

At the time the first set of dynamically balanced propellers were tried in one of the destroyers, the author, unfortunately, had no suitable instrument available for recording vibrations. It can only, therefore, be

asserted that, in the opinions of the ship's and dockyard officers and the author, there was a decided improvement. In the subsequent experiments on the "Town" class cruiser, the author was able to obtain a permanent record of the difference in vibration between two sets of propellers, the one set being tried on the ship as supplied by the makers, and the other after being dynamically balanced by ourselves. In order that the conditions for vibration might be the same, the loading of the vessel in the two conditions was maintained as similar as possible. Typical results, as graphically recorded, are shown in Figs. 1 and 2, and Fig. 3 shows the curves deduced from the records obtained. In the latter, the upper curve *A* shows the maximum readings obtained with the unbalanced propellers. It will be observed how, at the highest speeds reached, viz: about 493 revolutions, the vibrations increased at a rapid rate, whereas in the upper curves *BB* the maxima were not only much less, but of a different character. In the *A* curves, periods of relatively intense vibration, alternating with periods of comparatively small vibration, occurred at frequent but varying intervals, whereas in the curve *B* these periods of intense vibration had become as much reduced as to be almost absent, occurring only at long intervals.

In an appendix to the paper, the author gives illustrations of the balancing apparatus in use at the dockyard, also views of one of the cruiser's propellers, showing where weight required to be removed, together with a description of the general principles of the procedure to be adopted.—"The Shipbuilder."

STRENGTH OF MATERIALS.

The Heat Treatment of Large Forgings.

By Sir William Beardmore, Bart.

(*Institution of Mechanical Engineers; March,* 1917.)

In the heat treatment of large forgings there are no metallurgical principles involved which do not apply with equal force to the heat treatment of small forgings, such differences as exist being entirely due to the limitations which large forgings impose on the practical conditions under which the heat treatment is carried out. For every class of forging it is desired that the material from which it is made should give the mechanical tests required by a suitable choice of composition, but in addition it is most necessary that the material should be in such a physical condition that brittleness and the chance of sudden failure should be reduced to an absolute minimum.

In carbon-steel forgings produced to meet a definite mechanical test specification, this is equivalent to saying that the steel must possess the least crystalline growth or the smallest grain size, and the object of all heat treatment is to confer this condition on the forging before it leaves the steelworks. It is well known that, at any chosen temperature, the time the material is kept at its heat and the time taken to cool down to normal temperature again have an all-important influence on the grain size, and it is for this reason that in large forgings all the difficulties of heat treatment are magnified; and, in the author's opinion, a limit is ultimately reached in the size of the forging beyond which a plain carbon steel cannot be used with safety, and the use of an alloy steel becomes imperative.

The slowness with which heavy masses of steel cool down results in large crystal grains at the center of the mass, and, to avoid this, recourse has to be made to oil-quenching to hasten the cooling down, and so to

diminish the size of the grain, while a subsequent reheating is generally necessary to remove the hardness introduced by the quenching operation. With small forgings a simple annealing will put the material into the condition which will give satisfactory and safe results, but for large forgings some form of heat treatment is very necessary to get the best results.

It is a fact which is not sufficiently realized by engineers that in plain carbon steel the effect of oil-quenching is not uniform over the cross section, but diminishes the farther one goes from the outside surface. With very large forgings, therefore, a stage is reached at which, owing to the size of the forging, the quenching effect at the center is so small that it is insufficient to confer any benefit on the material, and it becomes impossible to guarantee the uniform results which are wanted.

The effect of work is very important, and there is an intimate relation under practical conditions between the grain condition of a forging and the amount of work which has been put on it during forging. For the very best results, the ratio of the cross section of the ingot and the largest cross section of the forging should have a minimum value, which it is advisable should not be less than three. Preferably it should be as great as possible, but it cannot always be adhered to, since for large sizes it is limited by the maximum size of ingot with which the presses can deal and sometimes by the shape of the forging itself. For an ingot 83 inches in diameter the maximum size of the forging should not exceed 48 inches for the best practice, and from a steelmaker's point of view, until the demand warrants the outlay of plant for dealing with still larger ingots, larger forgings than this should be built up rather than manufactured in one piece. In this connection the cause of commercial efficiency would be better served if engineers, in bringing out new designs in heavy steel construction, would consult the metallurgist and the steel manufacturer as to the best method of using their combined knowledge in the design and production of special requirements.

For high-tensile material—that is, material over 40 tons strength—where the factor of safety is limited owing to the special conditions under which the material has to be employed, it will generally be safer to use an alloy steel, since for the same tonnage a very much tougher material can be developed. But for material whose tensile strength is below this figure there is not the same necessity, and excellent results can be obtained from oil-treated low-carbon steel.

Fig. 1 indicates the outline of a large rotor spindle which was made to the following test specification:

Yield point	20 tons per sq. in.
Ultimate stress	36 tons per sq. in.
Elongation on 2 inches	24 per cent.

It will be seen from the drawing that the largest diameter was 3 feet 8¼ inches, and the minimum diameter only 11⅛ inches, so that if such an ingot were used as would give the requisite amount of work for the largest section, the amount of time and money spent in forging down this ingot to 11 inches would be out of all proportion; and so in this case, as in many other cases, the best metallurgical conditions are incompatible with the best commercial conditions, and a compromise has to be effected. An ingot 60 inches in diameter was used of the following composition:

C.	Mn.	Si.	S.	P.
0.38	0.89	0.21	0.029	0.044

In the heat treatment of this job, and of jobs having similar shapes, when the forging is put into the furnace the time taken to heat up the

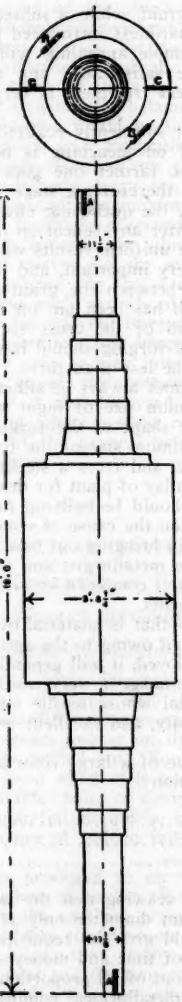


FIG. 1.—LARGE ROTOR SPINDLE.

heavy center section is much longer than the time taken to heat up the end sections. For instance, the end pieces of 11 inches diameter could be heated up in about four hours, while the center would take about 20 hours to reach the same heat, so that before the forging would be uniform the end pieces would have been at their heat for 16 hours, and would have

grown such a crystalline grain in consequence that any good effect of the oil treatment would have been annulled. To avoid this the ends were covered with asbestos sheets, leaving only the center exposed when the forging was charged into the furnace, and this sheeting was removed after 15½ hours in the furnace, so that the forging attained its heat uniformly as a whole, and the crystal grain in consequence was uniform throughout.

The test results obtained were very good, and are given in Table I. *A* refer to length tests taken one from each end of the forging, *B* to tests taken circumferentially from a ring taken from the center part after treatment, and *C* to tests taken radially from the same ring. In both *B* and *C* tests two tests were taken from opposite diameters.

TABLE I.

Fig. 1.	Yield point. Tons per sq. in.	Ultimate stress. Tons per sq. in.	Elongation (on 2 ins.) Per cent.
A.....	22.6	38.8	27
	22.8	39.2	27
B.....	22.8	39.2	28
	22.4	38.8	28
C.....	22.4	38.8	30
	22.4	35.8	24

The treatment which was given to this rotor was:

(1) Heated to 800 degrees C. (1,472 degrees F.) for two hours and cooled in oil. The center part attained its heat in 19 hours 30 minutes, and the ends were uncovered after 15 hours 30 minutes, so that they attained their heat simultaneously with the center part.

(2) Heated to 550 degrees C. (1,022 degrees F.) in 11 hours 30 minutes, and kept at heat for two hours.

A very troublesome feature of large forgings which have been heat-treated is the distortion which takes place, due to internal stresses probably set up by the operation of quenching. On machining the forging, and especially on removing the outside surface, these stresses are partly relieved, and the forging distorts and takes up a new shape under the influence of the forces still left in the material, so that it loses straightness and generally adds very largely to the time occupied in machining. It is advisable on this account to rough-machine the forging after treatment, and then to reanneal it before the final treatment is done. Allowance for this reannealing can be made on the first testing, and where the final machining is intricate and accurate it will be found to well repay the cost, since trouble in the machine shops due to distortion will be reduced to a minimum.—"The Shipbuilder."

LAMINATED GEARS.

The Laminated Gears, Ltd., of Sedgley Road, Sheffield, have favored us with some particulars of wheels introduced by them under the name of laminated gears. These are intended as a substitute for raw-hide and similar pinions, they are made up of a succession of steel discs—1/16 inch and upwards—which are assembled together to the required thickness cast, or in the case of small wheels on a turned mild-steel center or boss.

A special and novel feature is the staggering of the teeth, which is

obtained by turning every alternate disc half a space forward, thus virtually halving the pitch. There is between each disc a thin sheet of metal, varying in thickness according to requirements, thus leaving a clearance for the teeth to pass without loss of efficiency. It is claimed that the staggering of the teeth and building up of the gears produce the following three excellent qualities: (1) Silence, as the method of building up eliminates the metallic ring of solid steel gears. (2) The reduction of backlash to a minimum. (3) Improved lubrication, as the staggering of the teeth enables the wheels to retain the lubricant very much better than with ordinary gears, and instead of being squeezed out at the sides, it passes from space to space between the teeth round the wheel. Owing to their construction—it is further claimed—the efficiency of these gears is so much increased that a reduction in width of face from 15 per cent. to 20 per cent. can be made, as against that of ordinary gears. They may be applied wherever spur gearing is used, but are particularly recommended for high speeds where silence combined with accurate running without backlash is essential.—"Page's Engineering Weekly."

CORROSIVE ACTION OF SEA WATER.

When two metals are in contact with one another, and with sea water, a voltaic or galvanic action is set up, one of the metals becomes electro-positive and the other electro-negative, and consequently there is an eating away or corrosion of the electro-negative metal. This corrosive action of sea water is well known to shipbuilders and marine engineers, and the usual practice adopted to prevent it is to fix cast-zinc pads or rings near the bronze propellers or round the brass sea-cocks. The effect is to localize the voltaic action between the zinc and either of the other metals, and the zinc, being electro-negative, is eaten away, the other metals thus tending to escape corrosion. But the effect of the zinc is purely local, and it is found that severe pitting occurs at the top part of the inner surface of those pipes which are utilized to convey sea water to various parts of the ship. This is, however, not due to galvanic action, but due to the chemical action of an accumulation of hydrochloric acid formed as a result of the galvanic action. In the electro-chemical action between sea water and metals, both hydrogen gas and chlorine gas are evolved, and these two combine to form hydrochloric acid, which is a powerful solvent of metals. The gases rise and pass through the pipes when the valves are open, and accumulate at the top parts of the connecting pipes; hence the pitting at these places. It has been suggested that where *zinc rings are used, gas chambers should be fitted on the top parts of connecting pipes* as near as possible to the sea-valve, in order to trap the gases as soon as they are evolved, small pipes being fitted to the gas chambers to lead the gases through the hull above the water line.—"Shipbuilding and Shipping Record," June 21, 1917.

TWO AND ONE-QUARTER MILLION TONS OF MERCHANT SHIPPING BUILDING IN UNITED STATES.

A thorough canvass of the principal shipyards in the United States, made by "Marine Engineering" on May 1, shows that 2,250,000 tons of merchant ships are now under construction, 52.5 per cent. being built on the Atlantic and Gulf coasts, 33.2 per cent. on the Pacific Coast and 14.3 per cent. on the Great Lakes.

The report comprises returns from 96 shipyards. In all, 723 merchant

vessels of 2,248,000 gross tons are under construction. Three hundred and nineteen of these vessels, aggregating 1,539,354 gross tons, are freight steamships or motorships, 77 vessels of 365,488 gross tons are tankers, 10 vessels of 55,260 gross tons are passenger and freight steamers, and 118 vessels of 159,663 gross tons are wooden ships, either schooners or auxiliary schooners, while the remainder, or 199 vessels of 128,435 gross tons, are miscellaneous craft, such as barges, lighters, car floats, ferry-boats, etc.

The output of the various shipyards is shown in Tables I-V. A complete list of all the vessels under construction, showing the hull number, type of vessel, gross tonnage, builders and owners, is issued in printed form (price, 10 cents) and can be obtained by sending directly to the office of "Marine Engineering."

It is interesting to note that nearly 75 per cent. of the tonnage building on the Great Lakes is suitable for ocean-going service, while over 90 per cent. of the tonnage building on the sea coast is suitable for the deep sea carrying trade.

TABLE I.—FREIGHT STEAMSHIPS AND MOTORSHIPS.

	Number	Gross Tonnage
Albina Machine Works, Portland, Ore	6	21,800
American Shipbuilding Co., Cleveland, O	54	142,600
Ames Shipbuilding & Drydock Co., Seattle, Wash	12	105,600
Baltimore Dry Docks & Shipbuilding Co., Baltimore, Md	4	16,480
Bethlehem Steel Co. (Maryland Shipbuilding Plant), Sparrows Point, Md	10	61,184
Chester Shipbuilding Co., Ltd., Chester, Pa	8	43,600
Clooney Construction & Towing Co., Westlake, La	1	1,200
Columbia Shipbuilding Corporation, Portland, Ore	9	51,570
Wm. Cramp & Sons Ship & Engine Building Co., Philadelphia, Pa	4	26,160
M. M. Davis & Son, Solomons, Md	1	1,000
J. F. Duthie & Co., Seattle, Wash	7	40,110
W. & A. Fletcher Co., Hoboken, N. J	3	.....
Fore River Shipbuilding Corp., Quincy, Mass	7	51,900
Great Lakes Engineering Works, Detroit, Mich	29	116,200
Harlan & Hollingsworth Corporation, Wilmington, Del.	7	19,330
Manitowoc Shipbuilding & Dry Dock Co., Manitowoc, Wis	9	20,750
Mathews Shipbuilding Co., Hoquiam, Wash	4	5,800
Merrill-Stevens Co., Jacksonville, Fla	1	1,700
Moore & Scott Iron Works, San Francisco, Cal	7	63,500
Samuel L. Moore & Sons Corp., Elizabeth, N. J	2	5,230
Newport News Shipbuilding & Dry Dock Co., Newport News, Va	6	35,800
New York Shipbuilding Corp., Camden, N. J	16	115,200
Pacific-American Fisheries Co., Bellingham, Wash	6	12,300
Pennsylvania Shipbuilding Co., Philadelphia, Pa	8	62,800
Pusey & Jones Co., Wilmington, Del	7	28,000
Seaborn Shipbuilding Co., Tacoma, Wash	1	1,000
Seattle Construction & Dry Dock Co., Seattle, Wash	9	53,900
Skinner & Eddy Shipbuilding Corp., Seattle, Wash	10	57,120
I. L. Snow & Co., Rockland, Me	1	1,600
Standard Shipbuilding Co., New York	9	65,700
Staten Island Shipbuilding Co., West New Brighton, N. Y	8	16,000

	Number	Gross Tonnage
St. Helens Shipbuilding Co., St. Helens, Ore	2	1,950
Sun Shipbuilding Co., Philadelphia, Pa	8	53,200
Tampa Shipbuilding & Engineering Co., Tampa, Fla	2	7,000
Texas Co. (Bath Yard), New York	2	12,000
Toledo Shipbuilding Co., Toledo, O	12	31,100
Union Iron Works Co., San Francisco, Cal	18	117,320
Willamette Iron Works & Northwest Steel Co., Portland, Ore	8	70,000
Winslow Marine Railway & Shipbuilding Co., Winslow, Wash	1	1,250
Total	319	1,539,354

TABLE II.—TANKERS.

	Number	Gross Tonnage
American Shipbuilding Co., Cleveland, O	1	5,200
Baltimore Dry Docks & Shipbuilding Co., Baltimore, Md	5	16,250
Bethlehem Steel Co. (Maryland Shipbuilding Plant), Sparrows Point, Md	2	18,400
Chester Shipbuilding Co., Ltd., Chester, Pa	7	38,500
Wm. Cramp & Sons Ship & Engine Building Co., Philadelphia, Pa	2	14,200
W. & A. Fletcher Co., Hoboken, N. J	3	.....
Fore River Shipbuilding Corp., Quincy, Mass	4	26,712
Great Lakes Engineering Works, Detroit, Mich	1	2,200
Harlan & Hollingsworth Corp., Wilmington, Del	4	25,305
Merrill-Stevens Co., Jacksonville, Fla	3	1,600
Moore & Scott Iron Works, San Francisco, Cal	1	10,000
Newport News Shipbuilding & Dry Dock Co., Newport News, Va	5	43,673
New York Shipbuilding Corp., Camden, N. J	7	53,688
Pennsylvania Shipbuilding Co., Philadelphia, Pa	6	30,000
Riter-Conley Manufacturing Co. (Baltimore Yard), Pittsburgh, Pa	2	3,000
Skinner & Eddy Shipbuilding Corp., Seattle, Wash	3	19,800
Sun Shipbuilding Co., Philadelphia, Pa	2	14,200
Tank Ship Building Corp., Newburgh, N. Y	6	12,000
Texas Co. (Bath Yard), New York	2	12,000
Union Iron Works Co., San Francisco, Cal	11	18,760
Total	77	365,488

TABLE III.—PASSENGER AND FREIGHT STEAMERS.

	Number	Gross Tonnage
Wm. Cramp & Sons Ship & Engine Building Co., Philadelphia, Pa	5	33,400
Harlan & Hollingsworth Corp., Wilmington, Del	2	13,400
New York Shipbuilding Corp., Camden, N. J	2	8,000
James Rees & Sons Co., Pittsburgh, Pa	1	460
Total	10	55,260

TABLE IV.—WOODEN VESSELS: SCHOONERS AND AUXILIARY SCHOONERS.

	Number	Gross Tonnage
Atlantic Coast Co., Thomaston, Me	3	4,800
G. S. Baxter & Co., Jacksonville, Fla	3	3,300
Bean Shipbuilding Co., Camden, Me	3	3,600
Benicia Shipyards Co., Benicia, Cal	2	2,500
F. S. Bowker & Sons, Phippsburg, Me	2	1,250
Brunswick Marine Construction Corp., Brunswick, Ga	5	9,550
Brunswick Shipbuilding Co., Brunswick, Ga	1	750
Clooney Construction & Towing Co., Westlake, La	7	4,756
Francis Cobb Co., Rockland, Me	3	2,747
Columbia Engineering Works, Portland, Ore	1	300
G. G. Deering Co., Bath, Me	1	1,400
Chas. E. Fulton, San Pedro, Cal	2	3,600
Georgia Shipbuilding Co., Savannah, Ga	2	1,720
Geo. A. Gilchrist, Thomaston, Me	1	1,200
Gray's Harbor Shipbuilding Co., Aberdeen, Wash	6	14,400
Richard T. Green Co., Chelsea, Mass	1	1,300
Hanlon Dry Dock & Shipbuilding Co., Oakland, Cal	1	3,500
Hillyer-Spearing-Dunn Co., Jacksonville, Fla	1	780
Howland & Nelson, Beaumont, Tex	2	2,000
Kruse & Banks, North Bend, Ore	5	7,000
McEachern Ship Co., Astoria, Ore	7	12,200
Palmer Shipbuilding Co., Noank, Conn	4	10,000
Peninsula Shipbuilding Co., Portland, Ore	4	6,400
Percy & Small, Bath, Me	3	3,825
Henry Piaggio, Orange, Tex	5	10,400
Puget Sound Bridge & Dredging Co., Seattle, Wash	3	7,500
Read Brothers Co., Fall River, Mass	2	2,200
Savannah Engineering & Construction Co., Savannah, Ga	3	3,750
Sawyer Bros., Millbridge, Me	1	600
Schultze Shipyard, San Francisco, Cal	1	200
Seaborn Shipbuilding Co., Tacoma, Wash	3	5,700
Sharpstown Marine Railways Co., Sharpstown, Md	1	1,100
Slidell Shipbuilding Co., Slidell, La	4	4,600
Standifer-Clarkson Shipbuilding Co., No. Portland, Ore.	2	2,950
St. Helens Shipbuilding Co., St. Helens, Ore	2	3,270
W. F. Stone Shipyard, Oakland, Cal	5	4,350
Arthur D. Storey, Essex, Mass	5	1,165
Townsend Marine Railway & Construction Co., Boothbay Harbor, Me	3	3,400
E. J. Tull, Pocomoke City, Md	2	1,600
Washington Shipping Corporation, Seattle, Wash	4	10,000
Wilmington Iron Works, Wilmington, N. C	2	4,000
Total	118	159,663

TABLE V.—MISCELLANEOUS CRAFT: BARGES, LIGHTERS, CAR FLOATS, ETC.

	Number	Gross Tonnage
American Bridge Co., Pittsburgh, Pa	93	44,942
American Car & Foundry Co. (Jackton & Sharp Plant), Wilmington, Del	5	3,071
American Shipbuilding Co., Cleveland, O	2	700
Wm. H. Baldwin, New Baltimore, N. Y	2	199
Baltimore Dry Dock & Shipbuilding Co., Baltimore, Md.	1	145

	Number	Gross Tonnage
Bath Iron Works, Bath, Me	1	650
Clooney Construction & Towing Co., Westlake, La	11	10,500
Coastwise Shipping Co., Baltimore, Md	11	10,500
Cowles Shipyard Co., Buffalo, N. Y	2	135
M. M. Davis & Son., Solomons, Md	7	10,500
Dubuque Boat & Boiler Works, Dubuque, Ia	2	1,415
W. & A. Fletcher Co., Hoboken, N. J	13	.....
Great Lakes Towing Co., Cleveland, O	2	142
Hillyer-Spearing-Dunn Co., Jacksonville, Fla	2	4,800
Kelly-Spear Co., Bath, Me	3	5,212
B. P. Lanteri, Pittsburg, Cal	2	.....
Manitowoc Shipbuilding & Dry Dock Co., Manitowoc, Wis	2	600
McAteer Shipbuilding Co., Seattle, Wash	1	20,000
Mathis Yacht Building Co., Camden, N. J	7	.....
New York Shipbuilding Corp., Camden, N. J	2	1,530
Pusey & Jones Co., Wilmington, Del	5	3,100
James Rees & Sons Co., Pittsburgh, Pa	2	300
Rohde & Sons Co. Shipyard, Baltimore, Md	5	2,950
Southern Transportation Co., Chesapeake City, Md	2	2,000
Spedden Shipbuilding Co., Baltimore, Md	5	1,644
Texas Co., Bath, Me	2	.....
E. J. Tull, Pocomoke City, Md	4	2,400
Union Iron Works Co., San Francisco, Cal	3	1,000
Total	199	128,435

—From "Marine Engineering."

INTERNAL COMBUSTION ENGINEERING.

TREATMENT OF HYDROCARBON FUELS.*

By H. G. Chatain.

After many other types of carburetors had been tried and failed, during several years of the early development of the liquid-fuel engine, the jet carburetor was brought out. This device had for its basic principle of operation the issuing of one or more small streams of gasoline into a rapidly moving mass of air. The inertia of gasoline in conjunction with the moving body of air possessed sufficient force so that the liquid was, strictly speaking, mechanically broken up and turned from a liquid into what might be termed a fog.

The design of this apparently simple piece of apparatus is indeed very complicated—so much so that in over twenty years with innumerable intelligent investigators working on the problem, it is still, although practical, far from being a comparatively perfect piece of apparatus.

It is well known that the gas formed by the mixture of air and gasoline fog is as highly explosive as possible when it is composed of 15 parts of air to 1 part of gasoline. This is proportioned by weight at atmospheric pressure. Above or below this figure the carburetion is not so good. Explosion, it is true, may occur, but its effect becomes mediocre, and ob-

* Abstract from a paper presented at the meeting of the Metropolitan section of the Society of Automotive Engineers, New York City, May 17, 1917.

viously the power of the engine falls off. The disproportionate mixture may be carried to such a point that explosion does not occur at all.

RATIO VARIES WITH THE COMBUSTIBLE USED.

It is well known that the ratio 15 to 1 varies with the combustible used. For example, it is necessary to have 15 parts of air to 1 of gasoline, but only 9 parts of air to 1 part of ethyl alcohol, 7 parts of air to 1 part of methyl alcohol, and so on for the various fuels. Therefore, if the carburetion of the air by the fuel is to be as good as possible, it is necessary that the proportions mentioned be kept constant through the range of speed, load and combinations of the motor. The changes in speed and in load produce a radical varying effect upon the carburetor, as the varying amount of air aspirated decreases or increases the depression or drop in pressure at the gasoline jet. Compensating for this varying depression has never been solved in an entirely satisfactory manner.

The first question that has confronted the constructor has always been this curious feature of the carburetor; that is, to find a practical mean between gasoline and air at various speeds, loads and combinations. The operator cannot be asked to make running adjustments by hand, therefore it is necessary, first, to find a means to make permanent the proper ratio, and, second, to charge the motor with keeping this permanency of mixture.

It was at first thought that this ratio of 15 to 1 would necessarily stay constant, even with the variations in the functioning of the motor, and that the relative value of the depression caused by the aspiration of the motor would remain constant with relation to the gasoline and keep the proper proportions, the orifices admitting the air and gasoline to remain fixed.

This assumption was an error. Air is a gas and gasoline is a liquid; therefore the masses of the two bodies are not equal, and consequently the inertia of each is unequal. After the liquid is put in motion by the depression in the intake pipe, it has an inertia greater than that of air; therefore the liquid will continue to come out of the orifices a moment or two after the load has been thrown off or the speed of the motor reduced, while, on the contrary, the air issuing through the orifices in the vicinity of the gasoline jet ceases instantly, or nearly so, upon the reduction of speed or load of the motor.

The permanency of the ratio would exist without any special apparatus if we were dealing with two gases; for instance, air and illuminating gas or natural gas. The mixture of the two gases would have an inertia quite the same, and they would come out of the respective orifices in a fixed proportion, once they were adjusted for various speeds and loads of the motor.

The conclusion was arrived at that with this type of carburetor some device would have to be utilized to admit more or less air and thereby increase or decrease the depression in the intake pipe, so as to keep the ratio in the proper proportion. Hand regulation of the device mentioned was at first resorted to, but this was found to be impracticable, as one never knew exactly when the carburetor was adjusted so as to maintain the proper proportion of 15 to 1, with the net result that at times the engine ran properly, but oftener it did not.

It is probable that the market price of gasoline has been, up to the present year or so, comparatively so low as to discourage endeavor in the field of utilizing kerosene or heavy hydrocarbons such as fuel oil, various crude petroleum, tar oil, etc. However, conditions are changing and will change so as to make it imperative that some other combustible than gasoline be used. Since most lines of endeavor are conducted along the path of least resistance, it would seem that the next combustible to be

utilized will be kerosene. This brings us to a consideration of what has actually been accomplished in the utilization of kerosene.

DEVICES FOR CARBURETING WITH KEROSENE.

The demands to solve the problem of utilizing kerosene have resulted in certain distinct types of devices, which are as follows:

1. The commonly accepted type of gasoline-jet carburetor with heat applied in various ways: (*a*) To the fuel itself; (*b*) to the air; (*c*) heated jacket; (*d*) heated jacket plus heated intake pipe.

2. Devices that will partly burn the fuel before it is admitted to the intake pipe, forming a more or less fixed gas intermingling to a greater or less extent with the vapor of the combustible.

3. Devices that admit the combustible in the cylinder direct, the valves of the motor being so regulated as to cause a partial vacuum in the cylinder prior to the admission of the combustible.

4. Where the combustible is impinged, by means of a pump or otherwise, directly upon a hot surface in the interior of the cylinder or a chamber adjacent thereto.

5. Devices that raise the temperature of the combustible to the boiling point, producing vapor but in no sense of the word a fixed gas.

1. To criticize the endeavors formulative of the first group is a comparatively simple matter. Primarily, the forces manifest in the gasoline carburetor are absolutely insufficient to turn kerosene into a fog, or rather put it in a condition such that it would commingle with the oxygen in the cylinder and be readily inflammable. Secondly, the source of supply of the heat is inconstant and is difficult to regulate. Its action lags; that is to say, when a sudden demand is made upon the motor for increased combustible, the heat requisite is not available until the combustible is actually burnt, therefore sudden variations in load and speed cannot be well taken care of by this process.

In starting the motor it is necessary to derive heat from an exterior source. This is inconvenient, however, and for stationary engines, certain classes of boats, or other applications, where a skilled attendant is available, this type of gas-making device has had and is having success.

FIXED-GAS TYPE OF CARBURETOR.

2. The second type, or what might briefly be called the fixed-gas type, presents many attractive features. In starting all that is necessary is to ignite, by means of a wick, part of the combustible, no other preheating being necessary. With proper regulation of the amount of fuel actually ignited and relying upon the aspiration of the motor, a reasonably permanent gas can be formed and regulated under the adverse conditions under which we are working. The writer has operated a device of this character in extremely cold weather with gratifying results. We certainly may look forward to further developments.

3. A device utilizing this principle was experimented with to some extent by the writer, and although it was possible to start an engine cold, the cycle of events occurring in the cylinder was so divergent from normal, owing to the very late opening of the intake valve, that the experiments were discarded, and no great amount of work was done along these lines; also for the reason that the scheme introduced what appeared to be excessive complication.

4. This method of utilization of combustible is well known. It has the insurmountable fault that it is necessary to preheat the engine; this, it would appear, can be done in a practical manner only by means of some form of blowtorch, requiring several minutes to heat the parts sufficiently to make starting certain. If a multicylinder engine, such as the prevailing

types of the day, were used, the complication and trouble would make the scheme highly undesirable.

5. About the same criticism can be made for this scheme as for No. 4. It is hardly worthy of further investigation for our purpose. The idea has about all of the attendant disadvantages of the steam boiler with a few added.

In passing from the utilization of kerosene, the next logical step is to the very heavy hydrocarbons, and although there is still a great deal to be accomplished in this field, much has already been done.

For comparatively large powers and for diverse purposes the high-compression oil engine (an engine in which the compression goes to 500 pounds per square inch or thereabouts) is an accomplished fact, hundreds of thousands of horsepower being in daily use throughout the world. A brief description of its principle, but only a brief description, is in order.

The four-stroke cycle and the two-stroke cycle are used, and each has its advocates who claim their respective superiorities. Pure air is taken or admitted to the interior of the cylinder. This is compressed to 500 pounds per square inch or thereabouts, and at a point approximating the upper dead center of the piston or slightly before, fuel is injected in the form of a fog or fine spray, and upon coming in contact or commingling with the highly heated air in the interior of the cylinder, inflammation results. This obviously increases the temperature of the air, which in turn is expanded.

Up to the present time, of all the devices used in rendering the combustible fit for consumption, the accepted form is the air injector. The function of the injector is to receive a definite quantity of combustible per cycle, which is usually propelled to the injector by means of some form of pump. It is immaterial at what portion of the working cycle of the engine this oil is delivered to the injector, provided it goes there in the proper quantity and in synchronism with the rotative speed. The quantity is a function of the load on the engine. To this same injector device air is led from a pump or suitable container under pressures varying in practice usually from 800 to 1,300 pounds per square inch.

The injector is also equipped with a cam- or otherwise-actuated needle valve. This needle valve being opened on or slightly before the upper dead center of the piston is reached, permits the contents of the injector to come out in the form of fog or fine spray. Another part of the injector of great importance is known as the flame plate. Its function is to divert in the proper direction the incoming spray, or fog, from the injector. This is usually accomplished by one or several small holes of suitable size pointing in the proper direction.

PROPER DIRECTION OF THE INCOMING SPRAY.

This pointing in the proper direction brings up a very important feature in regard to utilizing combustible in this manner. It is essential that the entire combustion chamber be filled simultaneously or as nearly so as possible, with the combustible. If the flame plate was arranged to distribute the combustible in only one-half of the cylinder, the power and output of the engine would go down to an appreciable extent. In this respect I wish to emphasize that the high-compression engine is distinctly different from the engines utilizing fuel that has been admitted prior to the compression stroke. In this instance the compression stroke serves to a great extent to thoroughly mix the combustible with the air and presents a homogeneous mass at the time the inflammation is started. The development of the art is such that with the injector, as described and commonly used, it is possible to operate an engine at rotative speeds as high as 550 or thereabouts. Although fulfilling my theory in regard to great power to destroy the viscosity of the oil, the accepted type of injector today

operating a 1,000-pound pressure per square inch is still incapable of furnishing the necessary force to operate properly at speeds higher than previously stated.

The time necessary for the oil and air to go through the circuitous passages of the holes of the flame plate, all of which are apparently essential in the breaking up of the fuel, is excessive when we contemplate utilizing this device for high rotative speeds or in engines for automotive purposes.

It is essential in the design of an injector to so construct it that oil will make its way to the combustion chamber in advance of the injection air. The reason for this will be better understood when one considers that the injection air is expanded from 1,000 pounds per square inch down to 500 pounds, with attendant refrigerating effect, which tends to decrease the temperature of the immediate surrounding air and render combustion slower and less positive.

A number of attempts have been made to discard the injection air and resort to solid injection. Nothing of this character has ever been attempted in this country, at least, so I believe. An English firm of note, however, has apparently carried this line of investigation to a point where it is at least workable; although from reports on the subject it would appear that the economy of their engines is lower than is customary and that they do experience a somewhat smoky exhaust with attendant difficulties which result from imperfect combustion.

In regard to internal-combustion engines of the high-compression type there is one very important feature which apparently so far has not been given a great deal of thought; that is, with the most approved type of injector and all in excellent operative condition, it is still necessary to have some 40 to 50 per cent. excess air in the cylinder to produce perfect combustion. If the engine is overloaded, smoky exhaust is produced, meaning that the inflammation is not completed during the working stroke. Obviously this is a waste of fuel, not to mention the detrimental effects upon the mechanical parts of the engine. This is indeed a most important point, because if a greater percentage of the air which is taken in per cycle could be carbureted and utilized, the size of the engine could be reduced. I have long believed that a big step in advance is due and is coming when other means is devised for subdividing the incoming fuel into finer particles than is done at present, permitting each tiny particle to become thoroughly oxygenated in the shortest possible time.

It is possible that an entirely different shaped combustion chamber than the existing types should be used; that is, a combustion chamber that will be constructed in a shape to conform with the jet of incoming fuel.

PRINCIPLES OF HIGH-COMPRESSION OIL ENGINES.

I have been asked many times if it would not be possible to apply the existing principles of the high-compression oil engine to motors that would be suitable for automobiles and kindred purposes. Invariably the question has been answered, that the existing types so far are not applicable, but it must be frankly admitted that there is no really inherent reason why such a motor could not be built. It means carrying out in a proper manner the basic principles underlying the existing types, but it also means a careful study of all the mechanical parts, and a redesign along rational lines of the device now used for carbureting or breaking up the fuel and rendering it fit to be used in the motor.

When it becomes thoroughly understood that these motors can be made to operate for less than ½ pound of fuel per brake horsepower per hour and the price of existing fuels becomes higher, the proposed idea will come about and become a reality.

Some form of two-stroke-cycle engine would adapt itself better for this

class of service than the four-stroke cycle. The reason for this is that in the two-stroke-cycle engine of this category the compression pressures used, of 500 pounds per square inch and over, would be sufficient to counteract the inertia of the reciprocating parts with the result that the pressures on the piston pins and crank pins would always be downward or in the same direction. As this would be the case, the trouble of taking up bearings for quiet operation would be done away with. It would appear that it would be advisable to experiment and investigate along the lines of solid injection utilizing extremely high pressure, possibly as high as 6,000 pounds or 7,000 pounds to the square inch rather than go to the complication of utilizing air for this purpose, with its attendant multiple-stage compressor and paraphernalia. It is barely possible that experiments conducted along the line of especially preparing fuel prior to its use in the motor might materially assist in solving the problem.

THE SOLUTION OF THE FUEL PROBLEM.

The fuel question is a serious one at the present time, but it is my belief that it is but temporary. The path of least resistance will undoubtedly be the application and utilization of kerosene, not as attempted through the use of existing types of carburetors, but with a device distinctly constructed and embodying the requisites for the purpose.

When such a device is brought about it will permit the utilization of enormous quantities of this class of petroleum, which is now more or less a drug on the market.

The next step, and especially for commercial purposes, will be the utilization of the heavier category of petroleum and such other oils as tar oil and oils from the lignites, deposits of which in the great Northwest are almost limitless.

It should be borne in mind that in an injector the following qualifications are desirable:

First, the particles of oil should enter the cylinder prior to an injection of air.

Secondly, the breaking up of the fuel is better accomplished if the oil meets the injection air in some portion of the injector at right angles. A circuitous passage is obviously desirable, but its resistance should not be too great in cases where a great number of injections are required per minute, as this delays the delivery of the fuel to the heated air contained in the cylinder and also delays the total inflammation of the mass.—"Power."

SIX YEARS' DEVELOPMENT OF THE WERKSPOOR MARINE DIESEL ENGINE.*

By Thomas Orchard Lisle.

The Nederlandsche Fabriek van Wertuigen en Spoorwegmaterieel, of Amsterdam, more popularly known as the Werkspoor Company, have completed 37 marine engines for 24 sea-going mercantile ships, representing about 36,000 H.P., at the time of writing this paper, and were one of the first firms to build Diesel-type engines on an extensive scale. Throughout they have remained steadfast adherents of the four-stroke cycle, and have aimed at turning out an engine that can be repaired in any port where steam engines are repaired.

* Abstract of paper communicated to the American Society of Mechanical Engineers; March, 1917.

The *Vulcanus*, the first sea-going full-powered ship to be propelled by Werkspoor engines, afforded the firm their initial practical experience of marine operations, while hers were the first Diesel engines to have short pistons, crossheads and guides. The *Vulcanus* is a ship of 2,080 tons' displacement and 1,235 tons deadweight capacity, and is fitted with a six-cylinder, direct-reversible, Werkspoor crosshead engine of 450 B.H.P. at 160 r.p.m. At the beginning, difficulties were experienced partly through lack of experience of the ship's engineers and partly through her air compressors, but the engines gave very little trouble, wear and tear of the moving parts being very slight. On one occasion a high-pressure air reservoir burst while the engines were being overhauled at Amsterdam. Bad welding had left a fracture, and the top part of the compressed-air bottle blew off. The indirect cause was the accumulation of water in the air retainer, which had not drained owing to a cracked internal pipe. This mishap led to dispensing with the welded-type air bottle, and the suspension of high-pressure bottles with covers and valves downwards, thus effectually preventing any unknown accumulation of water.

For years the detachable cylinder-head design has been the standard practice with virtually every make of Diesel engine. At the cylinder head—just level with the combustion—where cooling is most needed, there are few or no cooling water passages. In the Werkspoor design adopted after experience with the *Vulcanus*, there is, however, a complete water passage round the sides, as well as on top, of the combusion chamber, while the overhauling of the pistons is greatly facilitated by the introduction of a patented device.

Radical changes in design were made with the engine of the tanker *Juno* in 1912. The frames for supporting the cylinders were entirely dispensed with, and forged-steel columns substituted, stiffening being obtained with, and lateral stresses absorbed by, diagonal steel rods. The load of tension that these rods have to take is over 130 tons per cylinder, or about 780 tons for the 16 rods of the whole engine. Although there are cast-iron rods at the back of the engine, they are not secured to the cylinders, but have a sliding-surface fit and carry the crosshead guides, while, being hollow, they are utilized as inter-coolers for the air compressors. They allow an up-and-down movement of the cylinders, but side pieces prevent lateral motion.

With the later models, the fuel-injection valve is offset from the center, and there is at least a space of 4 inches for cooling water everywhere in the cylinder head. Offsetting this valve, in contradiction to all theories, is proving entirely successful in actual practice. As the supporting webs of the cylinders occasionally cracked, they were dispensed with entirely in later designs, the shape of the cylinder head being improved at the same time.

With regard to piston cooling, Werkspoor-engined ships now use sea water for cooling the cylinders, pistons and bearings, as there are not sufficient troubles to necessitate a change to fresh-water cooling. With sea water below 100 degrees F. practically no scale is deposited, and when it is below 70 degrees F. it is as good as fresh water; but pistons and cylinders must allow of no pockets, for, once scale or dirt starts to deposit, it quickly accumulates and so causes local temperatures. The present Werkspoor piston-cooling system, illustrated in Fig. 1, consists of a jet of water forced up through clearance tubes, having no rubbing parts, but moving up and down with the piston to prevent splashing. The piston is never more than about half-full of water, the broken surface of the water being more efficient for cooling than a solid mass. By means of a simple sight system, a chokage of water supply at any point can be detected instantly.

So far as the Werkspoor Company are concerned, the trouble with air compressors is a matter of past history. In the air-compressor system now adopted, the link-and-beam operation motion is still retained. The high and middle stages are arranged side by side, each having its own piston rod, while the first stage is directly below and also has its own piston rod. In this way every piston can be separately dismounted. All valves are interchangeable.

As a precautionary measure against crankshaft failures the size of the bearings has been increased, and forced lubrication for the main bearings has been installed. One camshaft is provided for running ahead and one for running astern, the astern camshaft being driven by means of a spur

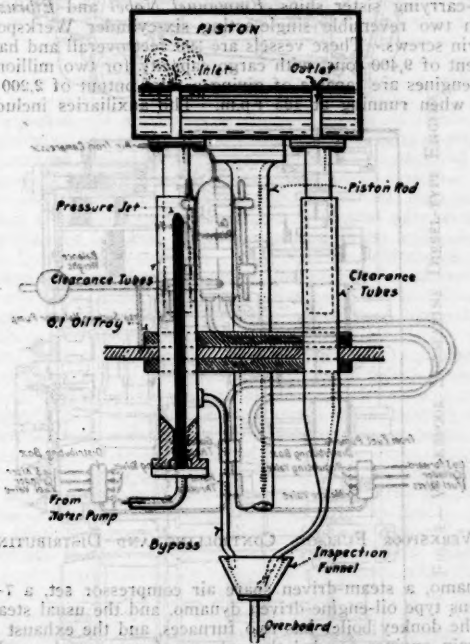


FIG. 1.—WERKSPOOR PISTON-COOLING SYSTEM.

gearing from the ahead camshaft. These two shafts are mounted on sliding brackets or saddles, which, when the vessel is maneuvering, slide to and fro and bring the ahead or astern cams under the rolls of the valve rockers. By making the rockers unusually large, the angle of the face of each roller, where it meets its cam, is never steep, and this produces smooth and silent running. The cams and cam rollers are of cast iron, and the steel tubes have a saw-cut down through the center, the two halves being held apart by cross-struts to eliminate vibration.

In the Werkspoor fuel-oil controlling and distributing system, illustrated in Fig. 2, a single pump, which supplies two distributing boxes, is fitted, each distributing box having feeds to three cylinders and every feed its

own adjuster. To the exhaust branch of each cylinder is fitted a thermometer, by which the engineer can instantly see if any one cylinder is receiving more or less fuel than the others. As a stand-by, an extra pump with a single change-over device is fitted. The fuel pump feeds a reservoir in which an air pressure is maintained; thence the oil passes through a governor-controlled throttle, a hand throttle, master valves, and finally into the distributors. The receiver holds enough fuel to run the engine for 15 minutes, so that in case of a breakdown of the fuel pump the spare pump can be connected up before the engine stops. The plunger pump is made from a solid steel forging; the valves are simply steel balls, and the gland runs in an oil bath.

The oil-carrying sister ships, *Emmanuel Nobel* and *Elbruz* * are each fitted with two reversible single-acting six-cylinder Werkspoor engines, driving twin screws. These vessels are 390 feet overall and have a loaded displacement of 9,400 tons, with cargo capacity for two million gallons of oil. The engines are capable of giving a total output of 2,200 B.H.P. for each ship when running at 125 r.p.m. The auxiliaries include a steam-

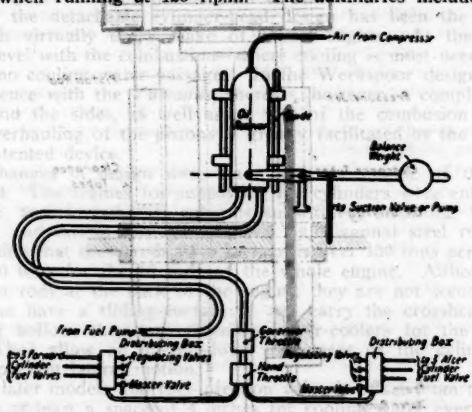


FIG. 2.—WERKSPOOR FUEL-OIL CONTROLLING AND DISTRIBUTING SYSTEM.

driven dynamo, a steam-driven spare air compressor set, a 7-H.P. semi-Diesel Brons type oil-engine-driven dynamo, and the usual steam-operated pumps. The donkey boiler has two furnaces, and the exhaust gases from the two main engines are led through one furnace and supply sufficient steam for operating the auxiliaries, including the steam steering gear. The other furnace, fitted with oil burners, is used at night, as more work being required of the dynamo, there is not so much steam available for steering. In the daytime at sea no oil-firing is required, for as much as 120 pounds' pressure can be maintained with the exhaust gases alone, while usually about 100 pounds is needed.

With the *Emmanuel Nobel* the recorded consumption of fuel for all purposes per 24-hour day is 9 tons of gas oil—8¼ tons for the main motors and ¾ ton for the boiler. This gives 0.29 pound per I.H.P. per hour under average running conditions. The total fuel consumption, including auxiliaries at night, works out at 0.32 pound per I.H.P. per hour. An

* A fully illustrated technical description of the *Elbruz* appeared in No. 46, Vol. X, of "The Shipbuilder."

oil-fired steamship of the same power, cargo capacity and speed (11½ knots) as the *Emmanuel Nobel* would have a total fuel consumption five times as great.

An important innovation, the reversing-blade propeller, has been fitted

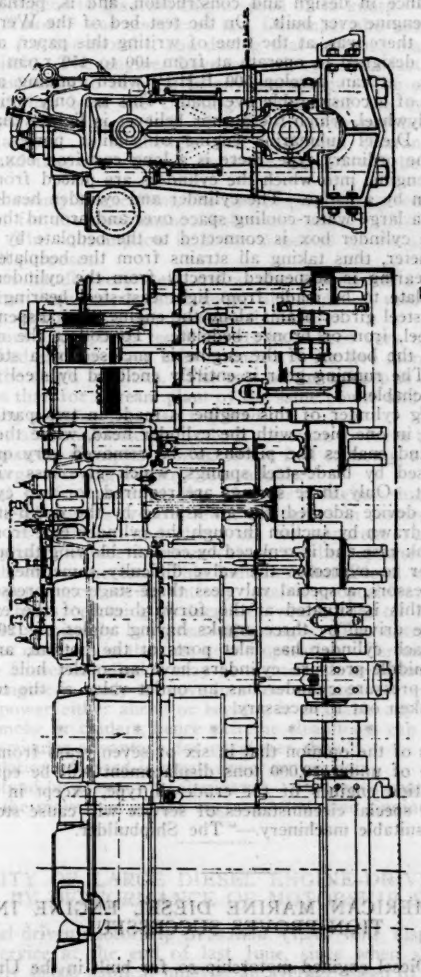


FIG. 4.—WERKSPOOR SUBMARINE DIESEL-TYPE ENGINE.

on four Werkspoor-engined vessels recently put into service. On these vessels power to reverse and actuate the mechanism is furnished by the engine itself. By interconnecting the throttle directly with the propeller mechanism, the strokes of the fuel pumps are automatically reduced and

increased by the mere act of throwing the blades into the central position or by reversing them. The advantages accruing from the adoption of this system are obvious.

Fig. 4 illustrates a new Werkspoor submarine motor that shows a remarkable advance in design and construction, and is, perhaps, the most simple Diesel engine ever built. On the test bed of the Werkspoor shops at Amsterdam there was, at the time of writing this paper, a six-cylinder submarine set designed to operate at from 400 to 450 r.p.m on the four-cycle principle. It can develop 500 B.H.P. when running at 400 r.p.m., and is capable of a considerable overload This set only weighs about 12 tons without flywheel, which the author believes is lighter than any existing two-cycle Diesel submarine engine of similar power. Instead of cylinders of the ordinary type, there is a long cast-iron box, running the length of the engine, into which the cylinders are placed from below and held in position by a flange. The cylinder and cylinder heads are in one, thus affording a large water-cooling space over and around the combustion chamber. The cylinder box is connected to the bedplate by steel rods 3 inches in diameter, thus taking all strains from the bedplate. In consequence each bearing is suspended directly from the cylinder, which enables the bedplate to be made from light cast-steel bearing blocks connected by cast-steel girder beams along the engine, thus dispensing entirely with a cast-steel, iron or bronze bedplate. To collect the oil from the working parts, the bottom of the engine is enclosed by a steel plate 1½ inches thick. The running gear is entirely enclosed by steel plates which are easily detachable.

Each working cylinder of this engine is made in two parts, the upper part being cast in one piece with the cylinder head, while the lower part is detachable and enables the pistons to be removed very quickly. The valves are closed by blade-steel springs, which give less vibration and take less height. Only three springs are required for two cylinders. In the air-cooling device adopted, the air aspired by the down-stroke of the main piston is drawn by suction through the cylinder box from the upper part of the crank case and is replaced by cool air blowing through the bent pipes. In order to overcome the valve difficulty sometimes experienced with air compressors, a special valveless three-stage compressor has been designed, and this is situated at the forward end of the engine. The three stages are driven by three cranks having angles of 120 degrees to each other. Each cylinder has inlet ports at the bottom, and the low-pressure and middle-pressure cylinders have an outlet hole at the end, while the high-pressure cylinder has an outlet valve at the top where it can be easily taken out if necessary.

* * * * * *

The author is of the opinion that in six or seven years from now every new cargo ship of under 10,000 tons displacement will be equipped with internal-combustion engines of the crude-oil type, except in a few rare instances where special circumstances of service will cause steam engines to be the only suitable machinery.—"The Shipbuilder."

LARGEST AMERICAN MARINE DIESEL ENGINE INSTALLATION PROVES SUCCESSFUL.

The largest Diesel engined motorship so far built in the United States is the twin-screw naval collier *Maumee*, of 14,500 tons displacement, propelled by two single-acting two-cycle Diesel engines of 2,600 shaft horsepower each. The engines were built by the machinery division of the New York Navy Yard, Brooklyn, N. Y., in accordance with original de-

signs secured from the Machinen Fabrik Augsburg Nurnberg, Germany, through the New London Ship & Engine Company, Groton, Conn. Each set of engines has six cylinders with a diameter of 25.2 inches and a stroke of 37.37 inches. At a normal speed of 130 revolutions per minute the engines are designed to give the ship a speed of 14 knots.

Since the *Maumee* was completed last December the vessel has been in active service for five months, and during that time has made 18,000 miles without a hitch. The engines have proved very satisfactory in service, and have been ready to respond to any emergency. The only defect that has developed has been the cracking of some of the cylinder jackets, which is apparently due to lack of sufficient clearance between working liners and jackets. It is thought that additional clearance can readily be allowed and that when this is done the engines will be ready for any service.

The fuel economy is excellent, the consumption being less than one-half that required for the most economical steam plants. Ordinary navy fuel oil is used. The consumption is about .5 pound of oil per shaft horsepower for all purposes.

The engines handle very well, and as the *Maumee* is a vessel engaged in service that requires repeated going alongside of vessels, docks, etc., the handling requirements are particularly severe.

The overhaul and upkeep requirements have been very meager, and but for the cracked castings have been attended to for five months' continuous running by the ship's facilities. The cost of the upkeep is apparently considerably less than for a steam plant of the same power and similar service. Less operating personnel is also required.

The weight of the installation is about the same as for a triple-expansion reciprocating steam plant, but, owing to the fuel economy, about 7 per cent. additional cargo-carrying capacity is available.

The actual performance of the *Maumee's* engines has demonstrated that large Diesel engines present fewer difficulties in service than do small or medium installations, and there are no apparent reasons why the power per cylinder on the *Maumee* (about 500 horsepower) should not be doubled with entire success. This indicates that Diesel engines of 5,000 horsepower per engine are an entirely feasible proposition.

For cargo boats operating in localities where fuel oil is available, the Diesel engine admittedly has great advantages. The extra first cost can very likely be made up in the first year's operation. Certain special advantages of the Diesel installation may be mentioned, as follows:

(*a*) The engine is ready to move in a few minutes.

(*b*) Full power either ahead or backing is available any time.

(*c*) No smoke or cinders, hence extreme cleanliness can easily be maintained (a particular advantage for passenger ships).

(*d*) Regulation of speed is much easier and certain.

(*e*) All stand-by losses are eliminated. Fuel consumption begins and ceases with the movement of the engines.—"Marine Engineering."

RELIABILITY OF LARGE DIESEL ENGINE-DRIVEN VESSELS SHOWN BY PERFORMANCE OF MOTORSHIP *SEBASTIAN*.

The Diesel-driven motorship *Sebastian* (7,200 tons displacement) was placed in service at the end of last June, since when she has covered approximately 40,000 nautical miles at an average speed of almost 9 knots, without a single breakdown, cylinder crack, or piston crack, cylinder-head crack, crankshaft fracture, etc., although she has been worked in the heaviest of Atlantic winter weathers.

The total number of engine stops at sea (accessory adjustments only) in

nearly ten months amount to four hours with one engine and five hours with the other engine, also six hours with one engine when salt water got into the lubricating oil and had to be cleaned out. The mileage covered would have been about 3,000 miles greater had she not been held up in England for fifteen days having certain alterations to her deck structure previous to her last recent voyage to New York.

Fuel consumption of the two main engines together has averaged under 6 tons (42 barrels) for 24 hours, with the engines averaging a fraction under 1,000 indicated horsepower *each* at 115 revolutions per minute. This figures out at 0.27 pound per indicated horsepower hour, or just over 0.4 pound per shaft horsepower. In her holds she carries 4,100 tons of cargo.

On the last voyage to an American port both her Diesel engines ran for sixteen days and nights without a single stop, voluntarily or otherwise, averaging 8.8 knots in most heavy seas and adverse winds.

In nearly ten months the main bearings have worn down 1/128th part of an inch. The crankpin bearings, although opened out for examination at regular periods, have only been readjusted once in ten months, the wear showing 0.2 millimeter. The crosshead bearings have been readjusted to 0.3 millimeter in the same period. Compressor piston rings are the same which came out with the ship, and show hardly any wear at all. A few of the main piston rings have been renewed, but this has been mostly due to breaking them when springing them off the pistons for cleaning purposes. The cooling pump valves are as good as the day they were put in, some ten months ago.

Furthermore, the Diesel engines, which were built by Werkspoor of Amsterdam, were not run or tested in the shops. They were built, sent to England in packing cases, and installed in the *Sebastian*, which was given a few hours' sea trial, and sent on her maiden voyage, arriving in New York in sixteen days.

The owners of the vessel are Lane & Macandrew, of London. All the engineers of this ship state that their work at sea is far less arduous than that aboard a steamer and that they have little to do other than watch the lubricating oil.

Diesel engines have enormous reserve emergency power, which can instantly be developed by increasing the pressure of the injection-air, and the chief engineer of the *Sebastian* considers that if chased by a submarine he could obtain for a few hours as much as 1,500 indicated horsepower per engine, although the builders' rating is 1,100 indicated horsepower at 130 revolutions per minute. Although passing through the war zone a motorship is nearly immune from submarine attacks because of the absence of smoke from her funnel, so that three miles away she is not visible to the periscope of a submarine in average weather, whereas the smoke of a steamer gives the U-boat about twenty miles' radius.—"Marine Engineering."

LOSS OF THE MOTORSHIP *SEBASTIAN*.

The following letter appeared in the "Shipping Illustrated" May 26:

"After a most successful career since her conversion to four-cycle Diesel power, the British oil tank motorship *Sebastian* was, as mentioned in your last issue, destroyed by fire and sunk in 200 feet of water off Nantucket Shoals while en route for Europe. Under no circumstances should this accident be used for the purpose of derogatory argument against the construction and use of Diesel motorships, for manifestly a motorship using crude-oil fuel without the use of exposed flame is far more safe and free of fire danger than is an oil-fired steamer.

"Since the war many Allied steamships have had mysterious fires while lying in American harbors, or shortly after leaving port, and likewise the recent burning of the tanker *Sebastian* is surrounded in mystery; but certainly the fire cannot have been due to the Diesel principle. The matter, however, is most regrettable, particularly as she has been a most successful vessel since her conversion, she having had no engine repair-bill.

"The writer has made every effort to secure an absolutely impartial report and has had interviews with both the chief engineer and the captain. It appears that the chief engineer, when in his cabin, noticed by the sound that one of the engines had stopped and that the other was slowing down. So he quickly went down below, and noticed that the fuel-oil had started to overflow from the filter, which is mounted on a grating over the engines, and the fuel-supply bottles that are suspended on a balance, had run dry (probably due to a chokage between the filter and the reservoir supply-bottles). The fuel-supply bottles running dry, of course, stopped and slowed down the engines respectively. Suddenly there was a tremendous flash of flame and the entire engine-room was on fire, the fire instantly spreading across in a big blaze.

"Now, the engine exhaust-pipes and the short extensions to each cylinder-head were lagged with asbestos, with the exception of three expansion joints on the exhaust-pipes and the flanges connecting the extensions to the cylinders.

"This fully indicates that the fire had absolutely nothing to do with the Diesel engines, because the exhaust temperature could not possibly have been sufficiently hot to ignite fuel-oil, and nowhere could there have been an exposed flame from the engines. To the writer it seems that there must have been some enemy plot that was arranged either before the ship left port or while the ship was at sea. Probably some highly inflammable chemical or gasoline was mixed with the fuel-oil before or after it was put in the bunkers. That the fire should have extended all over the engine-room adds to the mystery.

"The mean temperature of the exhaust gases of the *Sebastian's* engines under normal full-load conditions is about 700 degrees F., whereas when the accident occurred one engine had stopped owing to a stoppage of the fuel supply, and the other had slowed down, so that the exhaust-gas temperature must have been very low when the fire started. There was no electrical sparking device on the engines to start a fire.

"Mr. Arthur West, chief engineer, of the Bethlehem Steel Co., has since made some tests with a high-powered gas engine at their works. While the engine was running at a big overload and with high exhaust-temperature, fuel-oil of 33 degrees Baumé at 75 degrees F. (solar-oil was the residual fuel used by the *Sebastian*) was poured over the naked uncooled exhaust-pipe, close to the cylinders; but under no circumstances could they produce a flame—just a smoke, that's all! Kerosene then was tried with exactly the same results.

"Now this test was very severe, as it was made with an engine running at an overload and without an asbestos covered or water-cooled exhaust-pipe, so it is obvious that the residual-oil fuel known as 'solar-oil' running on to the exhaust-pipe, or any other part, of the *Sebastian's* engines could not have caused the fire and so supports my theory of a plot. No engineer of common sense would consequently blame the Diesel principle as the cause of the fire.

"However, the position of the fuel-filter on a grating over the engines was not a good one, and here it may be mentioned that the builders of the engines did not carry out the installation. The ship was not a new one, and the Diesel engines were fitted in rather a small space at the stern, which probably accounts for the position of the fuel filters, and they

should have been arranged elsewhere. Most of the old steam auxiliaries were retained, and at one end of the engine-room was an oil-fired donkey-boiler, but whether this boiler had anything to do with the fire I do not know.

"On her last voyage to New York, the *Sebastian's* engines both made a 16-day run without stopping, even for a few minutes, averaging 8.8 knots in heavy weather, although three of the assistants in the engine-room had never before been to sea in a motorship, which fact speaks for itself."

AERONAUTICS.

A FRENCH AVIATOR'S MESSAGE TO AMERICA.*

By Lieut. de la Grange, of the French Commission.

At this time, when you are preparing to make the greatest effort in aviation that has been made so far by any country, I would like to tell you why I have such confidence in your undertaking and what great results one has a right to expect from it.

I know that in a country where aviation, as a military arm, has been ignored until now, it seems inconceivable to certain people that such formidable amounts of money have been asked to carry out the program which your committee has mapped out. I am not surprised that some people have honestly asked themselves if it would not be a mistake to expend so great a part of the industrial and military effort of the United States in aviation.

It is you, gentlemen, because you have a better knowledge of the situation than anyone else, who will have to say again and again that no sacrifice will be in vain to secure the mastery of the air. You, yourselves, must be absolutely convinced of this, so that you in turn may convince others.

This war has been full of surprises. At first the Germans took the initiative, using methods which were convenient for them and suited their temperament; but during the short period when the Germans were compelled to fall back on the Marne, the war was directed by the Allies. Since then, however, the Germans, aided by their efficient methods—fruit of long years' labor—have given the war a defensive character. This is the reason why none of our offensive moves so far have been completely successful. Whenever the armies have met in the open the Allies have had the upper hand, but they have been unable to pursue the advantage, for, sooner or later, they were checked by that formidable array of all the arms that modern science has invented. Fortunately, there are two domains where, if they truly wish, the Allies can reign supreme: the sea and the air.

I will not talk of the naval campaign. That is not my specialty. But in all which concerns aerial warfare I am thoroughly convinced that we must work with supreme energy, and that there the United States can play a great part. In this direction the enemy can do nothing to bar our way.

The enemy may prefer to fight on land with mine and shell, burrowing themselves into the ground, rather than struggle in the open, in the good French way, facing each other. In the air, at least, they must fight man to man, or acknowledge themselves defeated. The air belongs to everyone, or rather to him who knows how to conquer it with a high hand. There traps do not exist, nor hidden machine guns, nor other obstacles

* Address delivered to the S. A. E. at Washington, June 26.

that warp the conditions of the fight. In the sky there are two men, armed alike, who fight like knights of old in their tournaments. It is the bravest and the quickest who wins, and when he has won, the sky, German as well as French, is his.

You understand the beauty of this struggle, in which all is beautiful, even death. Is there any death more beautiful and clean than that of the aviator, struck down in loyal combat, by a bullet? The French love the risk and the danger, they play the game well, but they hate treachery. It is the same with the American. Yes, when our pilots and their brave English comrades appeal to young America, they know that it is filled with courage and high ideals.

Those who give themselves up to the skies of France in 1918 will take part in a great battle, the prize of which will be the supremacy of the air. If we lose, our military operations will be hindered, and our towns, whose sons have given their lives in defending them, will be a constant prey to the destruction wrought by German aviators. If we win, the field will be cleared for our enterprises. Our armies will regain complete liberty of movement, while the German army will be deprived of its most precious source of information. Moreover, it will give us a chance to push the war into enemy territory. Our bombing squadrons will harass reserve forces continually, and attack all the important strategic points behind the lines, and the Germans will no longer be able to feel secure anywhere. America will come to know the difficulties of this plan, but they will not hinder her. The work has already begun, and before long you will teach the enemy the valor of your pilots and the excellence of your machinery. Germany does not believe that you will ever be a redoubtable enemy. Just as she was mistaken in 1914 when she talked of the "little English army," so now she is miscalculating the "little American aviation." Our allies, the English, showed them in good style that they were much mistaken. You Americans will do the same. Some years ago the Kaiser proclaimed that the future of Germany was on the sea. America can proclaim now that her future in this war is in the air.

CLASSES AND TYPES OF MACHINES REQUIRED.

Three different types of machines are used in modern warfare—the fighting, the reconnaissance, and the bombing machines. All of these except the fighting machines must be able to carry out light bombing; that is to say, they must be able to transport 200 pounds of bombs. When we say bombing machines, we mean a machine which is designed only for bombing purposes. It must have a carrying capacity of about two tons.

A fighting machine must have the following qualities: speed, handiness, great climbing ability, and high ceiling. The constructor must bear this in mind, and never sacrifice one of these qualities for the benefit of one of the others. If a machine was so built that it would be excellent for fighting at low altitudes, but would be no good above 12,000 feet, such a machine could not be adopted by the army, because the conditions of modern warfare are such that the machines are constantly obliged to seek high altitudes. It is not rare nowadays to meet German airplanes at a height above 18,000 feet. When it comes to fighting, the quality the pilot most prizes in his machine is handiness, of which speed is one factor. Therefore, you must not forget that you cannot value exactly the worth of a machine before it has been tested thoroughly by old and experienced pilots.

Our machines have always been, and are still, more handy than the German ones. This is one of the reasons why our fighters have been so successful.

The reconnaissance machines ought to have the same qualities as the fighting machines, but naturally they cannot have them to the same degree, because they have to carry more weight, and therefore they are bulkier.

They have to carry an observer, extra fuel, a wireless apparatus and, occasionally, bombs. The landing speed of these planes is a factor that must be taken into consideration for two reasons: the first is that as the best pilots will be used to fly the fast fighters, those that are left to fly the reconnaissance machines are often less skillful. The second reason is that night flying is becoming more and more necessary. The danger of landing at night must not, however, be exaggerated. It is merely a question of training.

The number of reconnaissance machines needed by an army depends upon the numerical strength of this army. It has proved useful to have four reconnaissance squadrons of fifteen machines each for every army corps of two divisions. But the number of both fighting and bombing machines is limited only by the possibilities of production, and by that of training a sufficient number of pilots. From a military point of view the reconnaissance machines are the most useful. However, these airplanes cannot carry out their work without the help of the fighting machines. The bombing machines are much less useful unless they operate in very large numbers, and in a sky where enemy machines are no more allowed to fly.

In theory, each airplane should be able to defend itself.

It must not be forgotten that speed and climbing ability will often be as good weapons as a machine gun. There is a saying in the cavalry that runs as follows: "The horse is the best weapon of a cavalryman." The aviator can say the same of his machine, therefore the pilot must have confidence in his machine, and the constructor must give the airplane a high factor of safety. As a rule, there is actually no danger in flying because of defects in the machine. The great danger comes from the high speed at which one must fly, which makes the slightest mistake made by a pilot when he is near the ground a fatal one.

All modern airplanes are of the tractor type. The reason is that in theory each airplane should defend itself. The pusher type, if attacked from the rear, could not defend itself.

TESTS.

Before an engine of a new model is accepted by the French army it undergoes a test of fifty hours. During five days it runs twice a day for five hours, during a half-hour of this time at full power, and for four hours and a half at nine-tenths of that power. For example, an engine which is said to produce 200 H.P. will be tested during five days twice a day for five hours, and will run for the first half-hour each time at 200 H.P., and for the following four hours and one-half at 180 H.P., this last to compensate the tenuity factor. An engine loses one-tenth of its power at 6,000 feet, because of the diminution of compression in the cylinders. Owing to the lessened pressure of the air, a smaller amount of gas is admitted in the cylinder at each stroke. When a new engine is finished it is tested for three hours on a testing block, then taken apart, examined, reassembled and tested for twenty minutes. The airplane when complete is accepted only after having been tested, and having fulfilled all the conditions required in the contract. The climbing speed is one of the most important factors.

DIFFICULTIES OF PRODUCTION.

The plane itself can be easily built; the same cannot be said of the engine. The airplane engine is very different from that of the automobile. The constructor is limited by the weight, and has to solve the difficult problem of designing an engine weighing only about two pounds per horsepower, which will be reliable and not cumbersome.

The automobile engine does not work constantly at its maximum power,

as the airplane engine does. Fortunately, at high altitudes the engine, as it gives less horsepower, is under less strain. It would be interesting to test the engines artificially, under different air pressures, so reproducing the conditions under which they will be used.

The constructors will have to use remarkably good material. The steel must have high resisting force and yet not be brittle.

The crankshaft and the roller bearings must be as light as possible to avoid vibration and allow the engine to turn at a high number of revolutions. Aluminum pistons have given very good results, because they are lighter, are excellent heat conductors, and therefore preignition is less likely. When alloyed with other metal, aluminum can be used for many parts of the engine.

Hardening is very important. It must penetrate progressively into the metal it is covering.

Lately most of the cylinders have been made of steel, but it is possible that cast iron may again come into use if we must build engines turning at 2,000 or even 2,500 r.p.m. The steel cylinder would not allow this, and though they are lighter the increased weight of the cast iron will be more than made up by the greater power of the engine.

Examining the different types of engines, we find that actually the qualities of rotary engines are balanced by those of the fixed engines. Each has its advantages, but it is probable that the fixed engine will soon be the only one which will be satisfactory, the weight of the fixed type, which was its great drawback, having been greatly diminished lately. It will probably be even more so in the future. Its horsepower can grow without its size increasing as much as that of the rotary engine. It does not seem possible to have a rotary engine which can turn faster than it does now, therefore, to increase its horsepower it would be necessary to increase so greatly the size of its cylinders that it would be impracticable. On the contrary, the fixed engine will certainly be brought up to 2,000 or 2,500 r.p.m.

AVIATORS.

It is almost impossible to say beforehand whether a man can become a good aviator. Some of the very best French pilots were in such bad physical condition that they had been rejected by the army. In spite of this, as it is necessary to train a great number of pilots in the shortest possible time, men in the best physical condition must be chosen. If after a short period of training they prove slow at learning, though not absolutely unfit, they must be rejected without hesitation. No time should be wasted over slow pupils.

An aviator must always do individual work, consequently he must have a great sense of duty. He must also be more intelligent and better educated than the average infantry man. The more mechanical and engineering knowledge the pilots possess the better it is. This they must be given during their training, so that when they are at the front they will be able to oversee the work of the mechanics and to test their engines themselves.

A good pilot is always busy with his machine when he is not flying. He must also be a good shot and practice daily. The importance of this is understood when we know that Guynemer, the "star" French pilot (who has brought down 43 German machines),* is now so skillful that he often brings down the machine he is attacking with a single shot.

MECHANICS.

The life of the pilot often is in the hands of his mechanic. The latter must be competent and careful. Each pilot in a squadron has two mechanics to take care of his machine, and one of them is an expert and does

*Guynemer has brought down his 50th machine at this writing.—Ed.

all the fine mechanical work while the other only cleans the engine, oils it and fills the tanks. The mechanics, as they always work for the same pilot, are proud of him and take the greatest care to keep his machine always in good order.

It is impossible to train a good mechanic rapidly, and, therefore, they must be chosen among the workmen who have a thorough knowledge of automobile engines. No important repairs are made at the squadron, so it is not necessary to have good adjusters there. The place for these is in the repair shops, where the engines are sent when there are important repairs to be made.

From these scattered remarks I hope that you can see what a powerful aid in aviation the United States can be to the Allies. Your men will make good pilots, your mechanics can be used at the front or in factories; and since your country is so great industrially, if the program which the American authorities have outlined is carried through, you ought to be one of the main factors in determining the strength of Allied aviation, which depends so largely on industrial excellence.—"Aviation."

THE STERLING-SUNBEAM-COATALEN 350-H.P. AIRCRAFT MOTOR.

The Sterling Engine Company, of Buffalo, has acquired the sole rights from the Sunbeam-Coatalen Co., of Wolverhampton, England, for the manufacture of the famous Sunbeam aircraft engines. As the company is in constant communication with Mr. Louis Coatalen, it is kept informed of changes as they are being made. All parts of the Sterling-Sunbeam motor are made in the United States and are so accurately made as to be interchangeable with the motors built in England and used by the Allies.

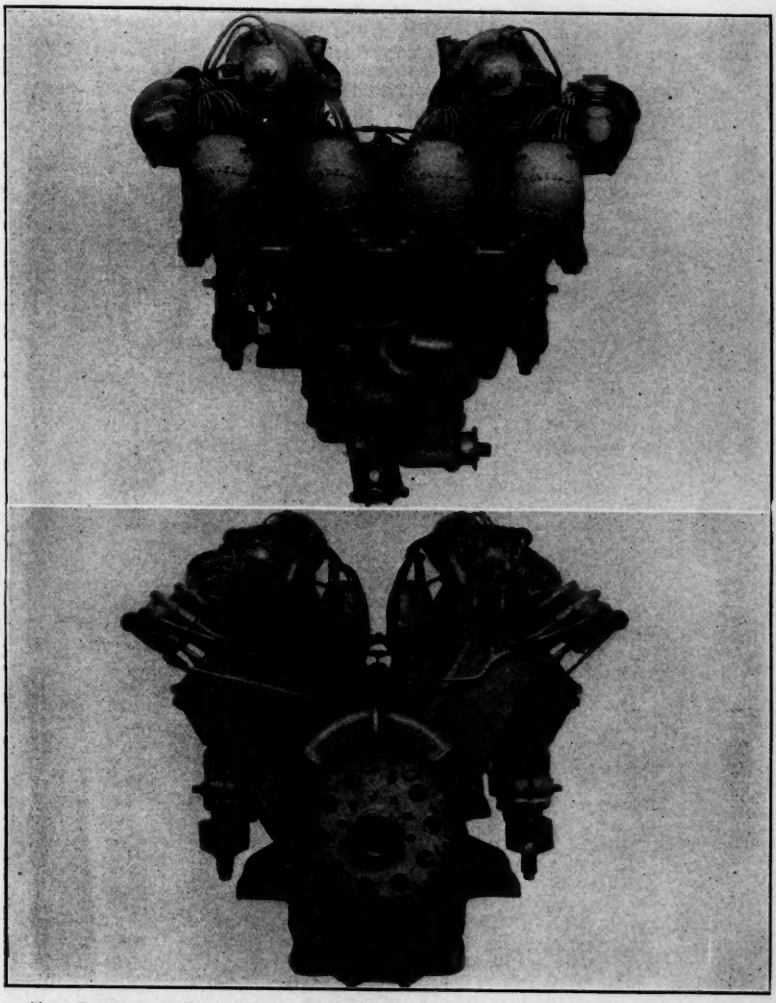
It has been learned from a reliable source that there are 16,000 of the 200-H.P. Sunbeam-Coatalen motors and 3,000 of the 350-H.P. motors in production in Great Britain. Another recent report has been received to the effect that a twin-motored machine, with two 350-H.P. Sunbeam motors made a flight over the Alps from London to Rome.

In handling the aircraft motor the Sterling Company has the benefit of years of experience in producing high-speed marine motors. Records established by Sterling-motored speed boats are interesting. The first eight-cylinder motor made in this country was made by the Sterling Company. The gold challenge cup was won for seven years by Sterling motors. In 1910 the *Courrier* won the first long-distance race at Palm Beach, covering 119.18 statute miles at a speed averaging 26 miles per hour. In 1911 the *Mitt 2d* won the challenge cup at Alexandria Bay. The *P. D. Q.* in 1912, the *Ankle Deep* in 1913, the *Baby Reliance* in 1914, *Miss Detroit* in 1915, and *Miss Minneapolis* in 1916, won the cup, covering 65.47 miles in an hour.

The Sterling Company is now prepared to supply the 350-H.P. engine in any quantity. The 200-H.P. motor, eight cylinders, weighs 475 pounds; the 350-H.P. motor, with twelve cylinders, weighs 1,058 pounds, and this latter is described here in detail.

The bore is 110 m.m. and the stroke 160 m.m. The gear ratio is two to one. The actual delivery of horsepower is 300 at 1,700 r.p.m., 332 at 1,900, 360 at 2,100, and 365 H.P. at 2,200 r.p.m. At 365 H.P. the propeller revolves at 1,100 r.p.m.

As in the foreign-made Sunbeam motors, castor oil has been found the best lubricant. The consumption of oil is .04 pint per brake horsepower hour. Fuel consumption, .49 to .5 pound per brake horsepower hour. The



THE LATEST 12-CYLINDER, 350-H.P. STERLING SUNBEAM AVIATION ENGINE.

gasoline used is ordinary commercial gasoline, about .62 Baumé. The compression ratio at sea level is 6 to 1. Mean effective pressure 135 pounds, which is within 15 per cent. of the theoretical possible mean effective pressure.

The crankcase is of aluminum alloy. The upper portion carries the bearings complete with caps. The lower half merely acts as a collector for the oil. The upper half weighs 87 pounds and the lower half 17 pounds.

The crankshaft is drop forged and counter balanced. Crankpins are hollow, plugged at each end to serve as oil leads. Bearings are babbit lined. Roller bearings are used on the shaft and a single-roll ball bearings for the thrust. The crank, complete with timing gear, weighs 102 pounds.

There are four camshafts, each driven by a set of spur gears. The camshafts are machined from the solid and heat treated. They are hollow and oil is fed through small holes to the campush rods. Weight of each camshaft 4½ pounds. The camshaft gear weighs one pound.

Connecting rods are articulated, with the auxiliary rod leading. The steel is produced in the United States from a European formula.

Pistons are cast from a special alloy. The inner wall is braced by webbing and the casting face scraped. Four piston rings are provided, three at the upper end and one at the lower. Weight of each piston, complete with rings, 2 pounds 3 ounces.

Each camshaft is covered in by a cast-aluminum housing, with removable inspection plates at each cylinder. Rocker fingers are interposed between cams and push rods.

Cylinders are of iron alloy, cast in blocks of three cylinders. Valve seats of cast iron and valves of Sunbeam-Coatalen special alloy. Each cylinder is provided with two intake and two exhaust valves.

Three oil pumps are located on the timing-gear end of the motor; two are set in front of tanks, which receive the oil through a filter and force it to the various parts of the motor, and the third pumps oil from the sump and sends it to the oil-cooling tank.

The circulating pump for the water is of aluminum. Its weight complete is 2½ pounds. The water enters the lower end of each bank of cylinders and emerges at the highest point. Provision is made for the employment of two flat, vertical radiators to be located on either side of the aeroplane fuselage.

Ignition is by four high-tension magnetos, two for each bank of cylinders, two "K. L. G." spark plugs for each cylinder. Up to the present this European plug is the only one which has stood up satisfactorily on the Sunbeam motor, but it is expected that a satisfactory American plug will be found. Each bank of three cylinders is served by a separate carbureter.

The propeller gear weighs 12 pounds and the reduction gear 6 pounds. The exhaust manifold weighs 10 pounds and the intake manifold 3½ pounds.

The propeller used in the tests at Buffalo was 12 feet 6 inches in diameter with an 8-foot 3-inch pitch. The propeller was made in England by the Sunbeam-Coatalen Co., and arrangements are being made to produce such propellers in this country. It is obvious that the best efficiency is obtained when motor manufacturers equip each motor with a propeller designed especially for it.

An unusually light, compact and effective air starter is provided for each set of six cylinders. This starter weighs only seven pounds complete with all valves and piping. The starters are built into the magneto end of the motor.—"Aerial Age Weekly."

AIRCRAFT AND MOTOR-CAR ENGINE DESIGN.

In a paper read before the Aeronautical Society of Great Britain on the 10th inst., Mr. Louis Coatalen said: "In some quarters the belief obtains that the design and manufacture of an aircraft engine is akin to that in use for a motor car; this proves to be, even on the most casual investigation, what old writers would have styled a vulgus." He proceeded to show that the two problems involved are fundamentally different. The chief characteristics of an internal-combustion engine for motor car service are:

(1) Weight is practically no object.

(2) Cost is of the utmost importance, therefore, there must be the minimum of machining, as instance the fact that the connecting rods of a motor-car engine are not milled, nor are the crankshafts machined all over.

(3) It must be capable of production in great quantities at minimum cost; otherwise, with the least amount of labor.

(4) It must be silent to the extreme of what is practicable.

(5) The maximum effort of which the engine is capable is not needed to be maintained for long at a spell. It seldom works at full power, and the brief duration of such effort explains the extraordinary reliability of even the inferior types of car engines.

(6) Flexibility, giving a constant torque at a crankshaft speed from 300 revolutions a minute to 2,000 revolutions a minute.

(7) Of course, this is a torque which corresponds to a very low mean effective pressure, namely, 80 pounds.

(8) The compression is relatively low and the valve area small, the cam forms being easy and the valve springs light.

(9) The system of lubrication, wherein the oil is carried in the base chamber of the crank case, suffices.

(10) The maximum horsepower required to be developed by any one engine rarely approaches 100. In the vast majority of cases it does not exceed 30.

By contrast he showed that the factors governing the design of an aircraft engine were in many ways distinctly different, and enumerated them as follows:

(1) Weight is of prime importance.

(2) Cost is not the deciding factor provided the necessary amount of power is obtained for the given overall dimensions of the engine, for its weight both as regards material and fuel, water and lubricant consumption; and that the desired degree of reliability is obtained.

(3) The amount of labor necessary to produce a satisfactory aircraft engine of high output is, and will be, always many times what is necessary in the case of a car engine, and is a matter of secondary importance provided the desired results are obtained. No machining is too expensive if it saves weight.

(4) Silence is relatively unimportant.

(5) The aircraft engine does all its work at practically full power.

(6) Flexibility, or evenness of torque, is of very secondary importance, because an aircraft engine is required to develop maximum torque at practically one speed only, or, at most, at an extremely narrow range of speed.

(7) But at its working range of speeds a very high brake mean effective pressure, say 130 pounds, is called for.

(8) The compression must be relatively high and valve area large, while the valve springs must be stronger than for a motor car, due to the cam form imposed.

(9) The high mean effective pressure necessary, coupled with the fact

of the engine doing nearly all its work at full power, involves a completely different point of departure in determining details of design, and, notably, the exploitation of new methods of achieving lubrication. Experience has demonstrated abundantly that when the base chamber is used as an oil well, as in motor-car practice, the lubricant soon becomes too hot, therefore, too fluid, resulting in reduction of pressure to the main bearings; hence the evolution of the dry-sump system for lubricating aircraft engines. Oil viscosity varies greatly with the temperature.

Such was his introduction in a highly instructive and informing paper, which should be carefully read and considered by those interested in both motor-car driving and aircraft flying. Concluding, he said:

"Assuredly it is interesting to compare present-day achievement with results of, say, three years ago, as instance those obtained at the Naval and Military Aeroplane Engine Competition in 1914. As a result, it will be found that then the use of aluminum was practically confined to the crank case only. Its application to the construction of pistons and other small parts of the aircraft engine was not known. Further, we find that the maximum mean effective pressure was approximately 106.5 pounds per square inch, and that the average fuel consumption was .6 pint per horsepower per hour. The weight of the engine with fuel for a run of six hours' duration varied from 9.55 in the case of the rotary air-cooled variety to 11.27 in that of the vertical water-cooled type.

By contrast, today the mean effective pressure standardized has been increased to 135 pounds per square inch, measured from the brake horsepower, and, in some cases, actually through the reduction gear. At the same time, fuel consumption has been reduced to .52 pint per horsepower per hour, while the weight of the V-type water-cooled engine has been brought down to 5.3 pounds per horsepower per hour with fuel and oil for a six hours' effort, all of which I hold to represent a notable rate of progress achieved in the brief period of less than three years.

From the inception of the movement several methods have been proposed for rating petrol engines. At the stage at which we have now arrived in constructing power plant for aircraft service some figure seems to be needed which will give a notion of the efficiency, or horsepower output, of an engine in relation to its size. Today the mean pressure is often used for this purpose; but in my view that is neither convenient nor can it be arrived at easily.

For this reason I wish to propose that the horsepower per unit capacity obtained from any given engine be taken as the standard for preparing the different "duties" of engines. Of course, the figure obtained is proportionate to the mean effective pressure, but doubtless it will be agreed that it is more convenient.

The capacity taken would be the capacity per cylinder multiplied by the number of cylinders and by the number of complete cycles per minute, but to serve the aim in view the horsepower per liter engine capacity per 1,000 cycles, otherwise per 2,000 crankshaft revolutions a minute, is proposed.—"Page's Engineering Weekly."

AMERICAN AEROPLANES AND MOTORS EQUAL TO EUROPE'S BEST.

People closely connected with the aeronautic movement and in a position to know what is being done in America and abroad have known that, with the possible exception of lack of finishing touches, some American aeroplanes and motors have been equal to the best European machines

and motors. Those that did not compare with Europe's best compared with Europe's second and third best—Europe having, as we have in America, constructors who do not keep up with progress and half trained and crank engineers who produce machines that look good on paper, but fail in actual performances. But that is to be found in every line of human endeavor.

There is printed elsewhere in this number the report of the Advisory Committee on Aeronautics which is coöperating with the Commandant of the Third Naval District in the organization of the Naval Reserve Forces, giving a detailed plan for the aerial defenses of the Third Naval District. This plan being basic, and adaptable for all the 15 naval districts, it was made a Congressional document. In this report the committee states that it has found that American aeroplanes are, type for type, equal to the best European machines. That report was a brief resumé of the following detailed report, which is based on a large mass of evidence.

COMPLETE REPORT ON THE EFFICIENCY OF AMERICAN AEROPLANES MADE BY THE ADVISORY COMMITTEE COÖPERATING IN THE ORGANIZATION OF THE NAVAL RESERVE FORCES.

The reports that have been made public heretofore alleging that American aeroplanes are not as good as European machines are based on conditions which existed in 1914-1915, at the time when American aeroplane manufacturers, like European manufacturers, were unable to get suitable metals for motors, and trained workmen to properly finish aeroplanes.

Before the war the demand for the best steel was supplied almost entirely by the Krupp agents, who, being subsidized by the German government, could underbid American, British and French steel makers and secure the orders. After the war started it was found that, contrary to the general impression, there was no Krupp steel in stock to be had. Therefore, experiments had to be made to develop steel equal to the Krupp steel, which was done after many months of experiments. In the meantime, the motor manufacturers not only in America, but in France and England, had difficulties due to soft and faulty steel.

With the exception of one firm, which is supplying England the large seaplanes of a type which is not obtainable elsewhere, and aeroplanes of the training type, which it has supplied as fast as the United States Government orders permitted, the British Government, which is the only government that has purchased aeroplanes in large quantities in the United States, has not placed orders for aeroplanes here.

AMERICAN AERONAUTICS HAS DEVELOPED TREMENDOUSLY SINCE 1915.

But since the summer of 1915, American aeronautics has developed tremendously, and our resources and sources of supplies of materials and skilled workmen have also developed tremendously. Likewise, the requirements for aeroplanes have changed, and an analysis shows that the change has been towards the lighter mode of construction, which was our mode of construction in 1915, but which was not approved by the European experts, who wanted extremely heavy construction.

In the summer of 1915, when the Curtiss Company, the Burgess Company and the Thomas Company took their first orders for machines for England, the British requirements were for machines of extremely heavy construction. That was the standard of the Royal Aircraft Factory at that time, and they wanted machines that equaled the heavy construction of the Royal Aircraft Factory. To give that strength involved cutting down the climbing and flying efficiency of the machine. Both the Thomas Company and the Burgess Company thought that some of the weight re-

quired by the British specifications was excessive, and built the machines lighter—and had the unpleasant experience of being told, when the machines were delivered, that the inspection committee considered them too light for use. Thereafter no further orders were placed with them. The Curtiss Company's machines were satisfactory, and after the trouble they had with motors in the summer of 1915, they have not had any further trouble and have continued to receive orders. Last summer an inspecting board held up the Curtiss orders to investigate the motor and find out whether there was a dangerous period of vibration in the motor. The investigation proved that there was not. Therefore, a new order was given to the Curtiss Company for motors.

AMERICAN SEAPLANES SUCCESSFUL IN ENGLAND.

The Curtiss Company's large seaplanes have been very successful and are being shipped as fast as the United States Government business permits. A report from a British authority was printed recently stating that they are rendering valuable service in patrolling the North Sea off the British coasts.

In the past eighteen months the war has forced a change in the requirements for aeroplanes, and the British specifications of eighteen months ago are considered dangerous today, because they make the machines too heavy and loggy. In a discussion that took place in the House of Commons on March 13, 1917, the following statement from a member of Parliament occurs:

"I would like to suggest to the honorable gentlemen on the front bench that the majority of these fatalities can be directly attributed to machines being sent up in which I said twelve months ago that our men were being murdered. I stand here twelve months later, and I say even on this very day that men are being murdered by being sent up in these machines. Our men are today being sent up in BE2 gun spotting. These machines are utterly incapable, armed and equipped as they are, of rising over 5,000 feet. They are sent up with two 20-pound bombs, and two guns fitted on them. The guns, however, are not adequately fitted for fighting. These machines are sent six miles out into the enemy's lines. Only recently I had two or three cases brought to my notice of men sent out gun-spotting, and with them they had two of what are called chasing machines, but they lost the chasing machines, they stopped over the lines at the mercy of the German airmen and they were eventually shot down. Quite recently, two fell in flames in our own lines."

We see by this statement that the British authorities are having trouble with the machines built on the Royal Aircraft Factory's specifications.

The report of the committee, which investigated the administration and command of the British Royal Flying Corps shows that this is only one of the many difficulties that have arisen in connection with the upbuilding of the splendid British air services. Progress was held back by red tape and slow action.

WAR HAS FORCED CHANGES AND CUT DOWN FACTOR OF SAFETY.

In the past eighteen months the war has forced a change in the design of aeroplanes, that is, to get greater efficiency and extreme performance, the weights had to be cut down. This, of course, involved greater refinement of detail and a very considerable reduction of the factor of safety. In fact, factors of safety as low as 3½ instead of the previous 6 to 8 are being accepted. Further, if these same machines were subjected to what is known as a "sand loading test" (the aeroplane is supported in an inverted position and sand or shot in bags is piled on the surfaces to approximate the load distribution in flight and increases made until

excessive elongation of parts, in case of tension members, or abnormal deflections are noted, or even rupture of any parts occur) it would be found that the factor of safety, thus determined, is far less than the calculated or theoretical factor of say 3½. Further, there is abundant evidence of actual failure of many of these machines in the air, resulting often in fatalities. Cases are on record, particularly with the English machines, of fuselages disrupting in the air under flight loads; tails actually collapsing and breaking off, and even the collapsing of wing structures.

ADVENT OF THE FOKKER BROUGHT RACING MACHINES INTO USE FOR FIGHTING.

The Germans, by bringing out the Fokker, brought into the war the type of small, fast machines, with high horsepower, which had been developed for racing in the years preceding the war. In 1912 and 1913 the International Aviation Trophy was won by aviators who made between 105 and 124 miles an hour, for a full hour, and over. In these races the aviators had gotten the utmost speed obtainable from such small machines without cutting the factor of safety down to a minimum. To beat the Fokker in speed and climbing ability it became necessary to cut down the strength of construction, and that has been done by both sides until today the aeroplanes of both sides are merely shells. That is true of the small fighting Nieuports, the Spads, and the Vickers on the Allies' side; and with the Fokkers, the Agos and the Halberstadts—the German "Spads"—on the German side. These machines, having wings of only 20 to 25 feet span, have not sufficient supporting surface to permit having a dominant advantage on either side. If the limited weight which the wings can lift, which is between 6 and 7 pounds per square foot, is put mostly in a large engine, as has been the case of late, which has resulted in equipping small machines with motors of from 120 to 160 H.P., the rest of the machine must be lightly built, cutting down the factor of safety to the danger point. The fuel carried has been cut down to the minimum requirement for a two-hour flight, and the machines are stripped of everything excepting what is absolutely necessary. Everything else having been reduced to a minimum, the only advantage obtainable is that afforded by the lightness or low fuel consumption. Many "light" air-cooled motors have excessive fuel consumption and are really "oil cooled."

PRESENT MOTORS ONLY NEED OVERHAULING ONCE IN 60 TO 90 HOURS — OVER 30 DIFFERENT TYPES BEING DEVELOPED IN AMERICA.

The general excellence of recent light-weight French motors is responsible for the good performance of the French fighting machines. The Germans have had good motors all along. The British motors are best in the larger types, above 200 H.P.

Now, however, motors in America are actually in acceptable production, which are the full equal of the best French motors to date. Further, it is of interest to note that a certain engine gives more power than its French prototype, of which it is an exact copy, and all parts interchangeable with the foreign built motor. Some American motors are as good as any abroad. Governor's Island Aero Squadron made about 2,500 flights in the past year with only about half a dozen cases of motor trouble. The motors used in the Mexican expedition gave no trouble whatever. Our present motors only need overhauling once in from 60 to 90 hours of flying and show no wear at the end of months of continuous use.

Further, let us record the fact that we have the talent available in America that can build machines and motors as satisfactorily as the experienced French and English designers and constructors, and can do the job on a real production basis. This is true of any size of aeroplane or

motor. There are now over 30 different types of motors being developed in America.

It may be put down as axiomatic in aeronautics that to get performance one must not see how much you can put on an aeroplane, but how much you can keep off. In other words, in the final analysis, it is low total weight and high horsepower, other things being equal, that gives remarkable performance.

In the struggle for the supremacy of the air a premium has been put on speed, and for the sake of speed machines are turned out with a factor of safety of only two or three. Of course, the factor of safety is of no value if the machine cannot outperform the enemy's machine. But if we will permit American manufacturers to adopt the European practice and cut down their factor of safety to two or three and only expect their machines to carry fuel for two hours, we surely can get aeroplanes at least equal to the European machines.

THE FOUR MAIN TYPES OF AEROPLANES IN WHICH AMERICAN EQUAL EUROPEAN.

But we know full well that so long as we have not got to fight off aviators from American skies no one will dare to assume the responsibility of cutting down the present factor of safety of American machines by half. In fact, a least factor of say 5 is highly desirable and will result in most satisfactory performance. They can fly the other types when they go to the front, and, as our safer types reach a speed of 120 miles an hour, they will be able to fly them with little additional training.

We who are close to developments and get a lot of information about what they are doing abroad as well as what is being done here, know that American aeronautic engineers can turn out, type for type, machines that are equal to the best European machines. To qualify this statement, there are four types of machines used in number abroad:

(1) *The Training Machines.*—There are half a dozen manufacturers who can turn out training machines equal to the very best European training machines.

(2) *The machine equipped with motors of between 120 and 200 H.P. to use for bomb dropping and for spotting artillery fire.*—After reading the account of the employment of this type of machine in the raid on the Mauser Works, written by one of the members of the Lafayette Corps who participated, I feel that we have the equivalent of these machines in the two-seater tractors being built by half a dozen manufacturers.

(3) *The large, slow, bomb carrier, which is used to carry bombs, but is protected during raids by the fast chasers.*—These machines are either single motored or twin motored. In the raid on the Mauser Works they were single motored. In other raids the twin motored have been used. The only reason we have not had many machines of this type in use is that no orders were placed for them until the close of 1916. Victor Carlstrom and Alan R. Hawley flew in the first twin-motored American battleplane from New York to Washington on May 26, 1916, and then the Aero Club of America offered it to the army, and it was sent to the Mexican border. The army accepted it in the beginning of June, but it was requested that changes be made in the landing chassis to meet certain conditions to be met at the Mexican border. The army officers were asked to have the changes made to suit their requirements, and in a letter received at the end of June they stated that they had placed an order for six machines of that kind, and as soon as they had decided on the type of landing chassis which they wanted they would have the changes made to conform with the other six machines. By October they had not yet settled on the type of landing chassis to be adopted.

But at the time of the flight we had occasion to talk to a prominent engineer, who had just returned from Europe, where he had seen all the European machines and reported that the Curtiss twin-motored battleplane was a more efficient flyer in every way than the best European twin-motored machines. This battleplane, equipped with two motors of from 160 to 200 H.P., can, we believe, equal any performances of European machines of the same type. Half a dozen other manufacturers, including the Aeromarine Plane & Motor Company, the Burgess Company, the Gallaudet Aircraft Corporation, the Sturtevant Company, the Thomas-Morse Company and other well established firms have received orders from the army and the navy, are now about to produce twin-motored hydroaeroplanes. With the experience they are gaining we can expect that they will soon turn out most efficient aeroplanes. This is also true of newer concerns, such as the New York Aero Construction Company, the A. S. Heinrich Corporation, the General Aeroplane Company and others. In some cases it may be necessary, in the interest of efficiency, to have these firms build on government specifications. But we may say that we have good prospects of getting machines comparable with the best European machines of the same type. Of course, we must order and use machines, as perfection can only be obtained through evolution by actual use. And we can only get production in quantity by placing orders for steady, continuous deliveries, as has been advocated by the Aero Club of America for the past three years and by the National Advisory Committee on Aeronautics at its recent meetings.

(4) *The fast single-passenger fighting machines.*—It is principally the fact that we have not had machines of this type in use that has created the impression that American machines are not as efficient as European machines. But the first demand for this machine in the United States came only a few months ago, and the first turned out show a speed of 122 miles an hour, with a slow landing speed of 55 miles; a climbing speed of 10,000 feet in ten minutes. And the factor of safety is up between 5 and 6. Of late the army and the navy have both been planning to place orders for machines of this type and half a dozen manufacturers are prepared to turn them out. The thing to be decided just now is whether the army and the navy are willing to adopt the foreign specifications and content themselves with fighting machines that have a factor of safety of two or three and carry two hours' fuel and land at a speed of close to 85 miles an hour. If they will content themselves with this, we feel certain that American manufacturers can turn out fighting aeroplanes that will make a speed of close to 135 miles an hour, which is the very best actual performance obtained abroad, and will have good flying qualities and fast climbing speeds.

But this low factor of safety and extreme landing speed, while necessary when it actually comes to fighting enemy aviators, can be dispensed with at a cost of only about five to ten miles per hour in speed, so long as our aviators do not have to fight enemy aviators and thereby save considerable injuries to men and machines during the period of necessary training. Our committee came to this conclusion after watching the succession of breakages of the fast British fighting machine which has been tested at Governor's Island during the past few weeks. This machine was so lightly constructed and landed at such great speed that there was breakage at most of its landings. Finally, two British officers, who brought it over, crated it away. The account of the death of Norman Prince, of the Lafayette Corps, tells that the machine he was flying landed at a speed of close to 100 miles an hour. The new Spad is almost as bad in so far as factor of safety and landing speed are concerned. These are machines to use in battle and American aviators will use them when they get there, but it would be wrong to say that they are better than the American fighting machines being turned out.

EXAGGERATED SPEED CLAIMS.

Recently there were printed in New York and Washington criticisms of slowness of American machines, coupled with claims of speed of 150 miles an hour for a newly arrived Vickers battleplane. A few days later those who followed the tests of this "battleplane" at Governor's Island witnessed a succession of breakages, due to the light construction and extreme landing speed of that machine. Again and again it was repaired, only to be damaged again on landing after a flight. The agents who brought it over finally put it away in despair. But even at that no claim was made of the machine having a speed of over 120 miles an hour, because it did not make it.

This brings us to the claims of excessive speed. Our committee has found that none of the reports of speeds of upwards of 130 miles an hour have been substantiated. Now, as it has been for the past ten years, we get reports of tremendous speeds attained, but find upon investigation that it is merely hearsay. In our official capacity we have had to consider hundreds of claims of records submitted for homologation, and usually found on actual measurement that the distance covered was much less than the distance claimed. The aviators usually based their figures on railroad distances, but did not allow for the fact that the aeroplane went in straight line, whereas the railroad had detours.

Miss Ruth Law, on her return from Europe, recently told our committee that the reports of performances of European machines, particularly the speed, were exaggerated, not intentionally, but as the result of the general habit of talking about great speeds. She is only one of many who have brought this to the attention of the committee.

A fifth type, the large seaplanes made almost exclusively in America, are admitted to be the most efficient for patrol and submarine hunting work. Within a year this type will probably be used in large quantities for launching torpedoes and to mount large guns.

The committee has also canvassed the field and finds that since the National Advisory Committee on Aeronautics has announced the plan of placing orders for aeroplanes and motors to extend over a period of three years, many large automobile and engineering firms have stated their willingness to take up the construction of aeroplanes and motors. So that, provided orders are placed immediately for continuous delivery, running over a period of three years, so that manufacturers can get and season wood and other raw materials and train workmen, the American aeronautic industry will be able to begin making substantial deliveries within ninety days and from 25 to 50 aeroplanes a day within five months.—"Flying."

AIRCRAFT THE REMEDY FOR U-BOAT MENACE.

In response to a request for a statement apropos of the third anniversary of the war, with special reference to the parts to be played therein by the United States and England, Viscount Alfred Milner, the widely known diplomat and statesman, indorsed most cordially the coöperation which America will render in the securing of aerial supremacy for our allies. He said:

"The thing that needs to be developed with the utmost energy—namely, aircraft, air services and air strategy. After many vicissitudes the Allies on the west front now have got something like the upper hand in the air, and by painful effort still continue to maintain it. But that is not enough. Complete supremacy in the air is certainly attainable through the united and maximum effort of the United States, France and Great Britain. They

should be satisfied with nothing less, for this alone would be decisive. Ten thousand additional aeroplanes may well be worth more than half a million or even a million men, and they are easier to transport. But here again everything depends upon realizing the extent of the need and to see that the task before us is as big as it really is. That is the first thing needful, and the second is like unto it—to hurry up and act in close concert with one another in all our preparations so that there may be no waste, no overlapping. Distance places many obstacles in the way of continuous consultation and pulling together. These are our greatest difficulties, but they can be greatly minimized by the proper organization of councils of the Allies."—"Aerial Age Weekly."

THE SUBMARINE.

BIBLIOGRAPHY OF THE LITERATURE OF SUBMARINES, MINES AND TORPEDOES.*

By David B. Rushmore,

Assisted by

William H. Lanman and Eric A. Lof.

This bibliography will appear in the August issue of the "General Electric Review." Advance publication is permitted by the courtesy of the magazine.

At the present moment there is a wide-spread desire for knowledge on the subject of submarines, the information concerning which is widely diffused and is difficult of location and access. Up to the present time but little has been known by the public in general on this subject, partly because of its apparent relative unimportance, and partly because much of the activities concerning these boats has been regarded as confidential and the subject somewhat shrouded in secrecy.

At the present moment submarines have an interest paramount to all else. They represent an advance made in the methods of naval warfare which has not yet been met by the development of a proper defence. Like the old race between the development of armor which cannot be pierced by any projectile and of projectiles which can pierce any armor, the development of methods of warfare has alternated between improvements in the offensive and defensive sides. Doubtless within a comparatively short period methods of defence against the submarine will be developed to such an extent as to render it comparatively harmless, but this has not as yet been attained.

A considerable effort has been made to gather together in one place the references on the subject of submarines to be found in books and periodicals, and while not absolutely complete may be considered as comparatively comprehensive.

It is hoped that the bibliography here published may be of some assistance in solving this important problem of the hour.

By far the best historical treatise to be had is the "Evolution of the Submarine Boat, Mine and Torpedo," by Commander Murray F. Suetter of the Royal Navy, published in 1908. A more recent book on the subject, of a somewhat "popular" nature, but containing much valuable information, is "Submarines, their Mechanism and Operation," by Frederick A. Talbot.

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*Mr. William H. Lanman, Engineer and Patent Attorney, 165 Broadway, New York City, has kindly offered to assist anyone who has difficulty in obtaining any desired book or article mentioned in this list, and it is suggested that those desiring further assistance on the subject correspond directly with him.

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Am. Soc. Naval Engrs. Jour., p. 158.

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List of Submarine and Allied Patents.

I.—Submarine Vessels.

Subject	Patentee	Patent No.	Date Issued
Submarine Boat	L. Alexander	7,609	Sept. 3, 1850
Submarine Vessel	L. D. Phillips	9,389	Nov. 9, 1852
Submarine Battery	J. Carver	41,365	Jan. 26, 1864
Submarine Vessel	J. Bachmann	44,380	Sept. 27, 1864
Submarine Torpedo Boat	S. S. Merriam	58,661	Oct. 9, 1866
Submarine Vessel	E. C. B. Rick	69,940	Oct. 15, 1867
Submarine Torpedo Boat	J. Jopling	173,018	Feb. 1, 1876
Torpedo Guards for Vessels	B. A. Richardson	193,727	July 31, 1877
Torpedo Boat	H. Mortensen	202,453	Apr. 16, 1878
Submarine Vessel	T. Nordenfelt	257,604	May 9, 1882
Submarine Boat	J. Jopling	273,851	Mar. 13, 1883
Torpedo Boat	B. B. Hotchkiss	278,708	June 5, 1883
Submarine Vessel	J. H. L. Tuck	297,647	Apr. 29, 1884
Submarine Torpedo Boat	P. B. Walker and H. G. Riggs	310,342	Jan. 6, 1885
Torpedo Boat	C. S. Lee	333,762	Jan. 5, 1886
Submarine Boat or Vessel	A. Campbell and J. Ash	344,718	June 29, 1886
Submarine Torpedo Boat	C. D. Goubet	352,124	Nov. 9, 1886
Submarine Torpedo Vessel	C. D. Shepard	379,991	Mar. 27, 1888
Submarine Vessel	C. De B. Shepard	379,992	Mar. 27, 1888
Submarine Vessel	G. Poore and W. C. Storey	399,693	Mar. 19, 1889
Submarine Boat	J. B. Gerber	406,725	July 9, 1889
Submarine Vessel	P. W. Pool	435,847	Sept. 2, 1890
Submarine Boat	J. F. Auer	470,535	Mar. 8, 1892
Submergible Torpedo Boat	J. P. Holland	472,670	Apr. 12, 1892
Submarine Boat	J. R. Haydon	493,266	Mar. 14, 1893
Submarine Boat	D. T. Freese and J. D. Gawn	521,854	June 26, 1894
Submarine Torpedo Boat	J. P. Holland	522,177	June 26, 1894
Submarine Torpedo Boat	G. C. Baker, dec'd	525,178	Aug. 28, 1894
Submarine Torpedo Boat	G. C. Baker, dec'd M. R. Baker Administratrix	525,179	Aug. 28, 1894

Submarine Torpedo Boat	G. C. Baker, dec'd M. R. Baker, Administratrix	530,466	Dec. 4, 1894
Submergible Boat	J. P. Holland	537,113	Apr. 9, 1895
Marine Vessel	S. Lacavalerie	538,921	May 7, 1895
Submarine Vessel	J. Scheubeck	575,907	Jan. 26, 1897
Submarine Vessel	S. Lake	581,213	Apr. 20, 1897
Submarine Boat	G. W. Bennum	604,185	May 17, 1898
Submarine Boat	R. Lincoln	615,866	Dec. 13, 1898
Torpedo Attachment for Boats	J. P. S. Sadtler	625,851	May 30, 1899
Submarine Vessel	S. Lake	638,342	Dec. 5, 1899
Submarine Boat	J. P. Holland	681,221	Aug. 27, 1901
Submarine Vessel	J. P. Holland	681,222	Aug. 27, 1901
Submarine Boat	J. P. Holland	683,400	Sept. 24, 1901
Visual Indicator for Submergible Boats	C. A. Morris	685,164	Oct. 22, 1901
Torpedo Boat	T. J. Moriarty	691,160	Jan. 14, 1902
Automatic Diving Mechanism for Submarine Boats	J. P. Holland	693,272	Feb. 11, 1902
Means for Automatically Ballasting Submarine Boats	J. P. Holland	694,153	Feb. 25, 1902
Submarine Boat or Vessel	J. P. Holland	694,154	Feb. 25, 1902
Submarine Boat	J. P. Holland	694,643	Mar. 4, 1902
Submarine Boat	S. Lake	695,215	Mar. 11, 1902
Submarine Boat	J. P. Holland	696,972	Apr. 8, 1902
Submarine Boat	J. P. Holland	702,728	June 17, 1902
Submarine Boat	J. P. Holland	702,729	June 17, 1902
Submarine Boat	J. P. Holland	706,561	Aug. 12, 1902
Submarine Boat	J. P. Holland	708,553	Sept. 9, 1902
Pneumatic Torpedo Firing Apparatus	S. Lake	709,335	Sept. 16, 1902
Storage of Supplies for Submarine Vessels	S. Lake	714,921	Dec. 2, 1902
Anchor Hoisting Device for Submarine Boats	S. Lake	716,059	Dec. 16, 1902
Submarine Boat	S. Lake	717,101	Dec. 30, 1902
Submarine Boat	C. B. Gillette	718,450	Jan. 13, 1903
Submarine Boat	S. Lake	726,227	Apr. 21, 1903
Submarine Boat	S. Lake	726,705	Apr. 28, 1903
Combined Ventilating and Observing Tube for Submarine Boats	S. Lake	726,947	May 5, 1903
Submarine Boat Structure	H. N. Ridgway	731,500	June 23, 1903
Automatic Diving Mechanism for Submarine Boats	F. W. Brady	738,879	Sept. 15, 1903
Submarine Boat	L. Y. Spear	739,734	Sept. 22, 1903
Submarine Boat	D. F. Toomey	746,606	Dec. 8, 1903
Tank for liquid fuel	F. T. Cable and L. Y. Spear	751,609	Feb. 9, 1904
Ballast Compartment for Submarine Boats	S. Lake	754,222	Mar. 8, 1904
Submarine Boat	S. Lake	756,030	Mar. 29, 1904
Buoyancy Regulating Apparatus for Submarine Boats	L. Y. Spear	772,970	Oct. 25, 1904
Compensating Device for Submarine Boats	L. Y. Spear	778,339	Dec. 27, 1904
Compensating Device for Submarine Boats	F. T. Cable	778,350	Dec. 27, 1904
Means for Submerging Submarine Boats	R. L. D'Equevilley	788,525	May 2, 1905

Subject	Patentee	Patent No.	Date Issued
Submarine Boat	R. D'Equevilley	798,501	Aug. 29, 1905
Submarine Torpedo Boat	J. J. Harpain	801,101	Sept. 19, 1905
Ballast Device for Submarine Vessels	S. Lake	803,174	Oct. 31, 1905
Ballast Apparatus for Submarine Vessels	S. Lake	803,175	Oct. 31, 1905
Air Supply Apparatus for Submarine Vessels	S. Lake	803,176	Oct. 31, 1905
Engine Exhaust for Submarine Vessels	S. Lake	803,177	Oct. 31, 1905
Submarine Vessel	S. Lake	803,178	Oct. 31, 1905
Buoyancy Regulating Apparatus for Submarine Boats	L. Y. Spear	805,496	Nov. 28, 1905
Submarine Boat	H. O. Eiane	803,885	Nov. 7, 1905
Construction of Submarine Boats	T. H. Wheless	811,886	Feb. 6, 1906
Means for Steering Submarine and Submergible Boats	L. Y. Spear	812,306	Feb. 13, 1906
Submarine Vessel and Mechanism connected therewith	G. Simpson	812,956	Feb. 20, 1906
Submarine Boat	J. P. Holland	815,350	Mar. 20, 1906
Submarine Boat	M. Naletoff	817,130	Apr. 3, 1906
Automatic Safety Depth Regulating Valve for Submarine or Submergible Boats	L. Y. Spear and T. S. Bailey	820,372	May 8, 1906
Submarine Boat	E. Miehoff	821,595	May 22, 1906
Duct Keel for Submarine Boats	L. Y. Spear	821,895	May 29, 1906
Submarine Boat	T. H. Wheless	822,565	June 5, 1906
Submarine Vessel	S. Wiebe	832,646	Oct. 9, 1906
Immersion Apparatus for Submarine Boats	E. A. Nilsen	834,161	Oct. 23, 1906
Vessel Construction	O. S. Pulliam	836,892	Nov. 27, 1906
Propulsion of Submarine Vessels	S. Lake	846,417	Mar. 5, 1907
Submarine or Submergible Boat	T. S. Bailey and L. Y. Spear	848,872	Apr. 2, 1907
Submarine Boat	J. J. Harpain	850,831	Apr. 16, 1907
Submarine Boat	L. Y. Spear	854,004	May 21, 1907
Submarine Boat	J. M. Cage	854,146	May 21, 1907
Submarine Boat	J. M. Cage	860,126	July 16, 1907
Submarine Vessel	A. Hector	863,532	Aug. 13, 1907
Submergible Vessel	E. L. Peacock	872,842	Dec. 3, 1907
Torpedo Boat	S. Lake	876,564	Jan. 14, 1908
Twin Screw Submarine Boat	L. Y. Spear	878,752	Feb. 11, 1908
Submarine Boat	G. Behrmann	896,613	Aug. 18, 1908
Shelter for Use in Connection with Submarine Vessels	M. A. Laubeuf	921,125	May 11, 1909
Submarine Boat	L. Y. Spear	922,056	May 18, 1909
Submarine or Submergible Boat	C. Laurenti	922,298	May 18, 1909
Submarine or Submergible Boat	S. Lake and E. L. Peacock	926,007	June 22, 1909
Submarine Vessel	S. Lake	926,065	June 22, 1909
Entrance Shaft for Submarine Boats	R. D'Equevilley-Montjustin	932,379	Aug. 24, 1909

Craft Adapted to Travel in a Resisting Medium	C. M. Stanley	939,344	Nov. 9, 1909
Submarine Boat	E. D'Equevilley-Montjustin	943,604	Dec. 14, 1909
Submarine Boat	R. D'Equevilley-Montjustin	943,605	Dec. 14, 1909
Propulsion of Submarine Boats	C. Del. Proposto	944,776	Dec. 28, 1909
Boat Salvage Device	G. Salles	945,141	Jan. 4, 1910
Torpedo Boat	H. Maxim	946,944	Jan. 18, 1910
Means for Expelling the Gas Engine Exhaust of Submarine Boats	L. Noe	953,283	Mar. 29, 1910
Method of Maintaining Equality of Weight in Submarine Boats while in operation	P. Winland	953,881	Apr. 5, 1910
Submarine Vessel	A. J. Griffin	958,742	May 24, 1910
Means for Emptying Submergence Tanks in Submergible Boats	L. Y. Spear	964,943	July 19, 1910
Submarine Boat or other Vessel	A. H. Atterbridge	969,128	Aug. 30, 1910
Submarine Boat	S. Lake	970,064	Sept. 13, 1910
Semisubmerged Submarine Gunboat and Torpedo Boat	H. Hertsberg, A. A. Low and M. J. Wohl	971,676	Oct. 4, 1910
Submarine Torpedo Boat	A. M. Fuller	970,210	Sept. 13, 1910
Submarine Boat Equipped with Submerging Planes	L. Y. Spear	973,227	Oct. 18, 1910
Submarine Vessel	S. Lake	947,921	Feb. 1, 1910
Device for Regulating the Submersion of Submarine Boats and the like	M. F. Hay	977,951	Dec. 6, 1910
Submarine or Submergible Boat	C. Laurenti	985,911	Mar. 7, 1911
Submarine Boat	R. D'Equevilley	988,632	Apr. 4, 1911
Submarine Vessel	M. A. Laubeuf	989,371	Apr. 11, 1911
Compensating for Torpedoes Discharged from Submarine Boats	L. Y. Spear	997,713	July 11, 1911
Submarine Boat	S. C. Rockman	998,204	July 18, 1911
Submarine or Submergible Boat	E. L. Peacock	1,003,063	Sept. 12, 1911
Ventilating Apparatus for Submarine Vessels	H. Hertsberg, A. A. Low and M. J. Wohl	1,006,380	Oct. 17, 1911
Conning Tower for Submarine and Diving Vessels	G. J. K. Behrmann and E. Hurlbrink	1,011,012	Dec. 5, 1911
Torpedo Tube Cap	H. Hertsberg and M. J. Wohl	1,006,212	Oct. 17, 1911
Torpedo Boat	H. Maxim	1,028,473	June 4, 1912
Submarine Boat	A. Ehrmann	1,014,959	Jan. 16, 1912
Safety Weight for Submarine Vessels	M. A. Laubeuf	1,035,021	Aug. 6, 1912
Hull for Submarine Boats	V. Cavallini	1,044,489	Nov. 19, 1912
Pressure Resisting Receptacle	J. Buttgen	1,045,673	Nov. 26, 1912
Rudders for the Submersion and the Navigation of Submarine Vessels Under Water	C. Laurenti	1,061,088	May 6, 1913
Submarine or Submergible Torpedo Boat	E. L. Peacock	1,067,371	July 15, 1913
Submergible Boat Construction	L. Y. Spear and H. E. Grieshaber	1,072,393	Sept. 2, 1913
Construction and Conning Equipment for Submergible Boats	L. Y. Spear and H. E. Grieshaber	1,072,392	Sept. 2, 1913
Submarine Boat	F. Fenaux	1,089,543	Mar. 10, 1914

Subject	Patentee	Patent No.	Date Issued
Air Supply Apparatus for Submarine Vessels	P. Parnitzky	1,097,356	May 19, 1914
Housing for the Conning Towers, Periscopes and Ventilating Shafts of Submarines	M. F. Hay	1,099,126	June 2, 1914
Periscope	J. Hanson	1,102,046	June 30, 1914
Submarine or Submergible Boat	M. F. Hay and F. Gurhauer	1,107,942	Aug. 18, 1914
Submergible Boat	G. M. Lagergren	1,108,192	Aug. 25, 1914
Diving Chamber for Submarine Operations	C. Petit	1,109,145	Sept. 1, 1914
Periscope for Submarine and Submergible Craft	B. Rosenbaum	1,110,827	Sept. 15, 1914
Torpedo Pilot Boat for Automobile Torpedoes	S. Danenhower	1,111,139	Sept. 22, 1914
Submarine Boat	H. Techel	1,121,210	Dec. 15, 1914
Pressure Regulating Air Circulating for Submarines	T. J. P. Aanstoots	1,115,367	Oct. 27, 1914
Submarine Vessel	C. Laurenti	1,120,392	Dec. 8, 1914
Buoyancy Regulating Apparatus for Submarine Boats	S. Lake	1,123,762	Jan. 5, 1915
Diving Rudder for Submarine Vessels	C. Laurenti	1,125,567	Jan. 19, 1915
Submarine Vessel	G. G. Ullin	1,125,772	Jan. 19, 1915
Submarine	J. M. Cage	1,126,616	Jan. 26, 1915
Submarine Boat	H. S. Epes	1,126,624	Jan. 26, 1915
Ballast Controlling Apparatus	S. Lake	1,127,648	Feb. 9, 1915
Shallow Draft Submarine Boat	F. B. Whitney	1,127,707	Feb. 9, 1915
Pressure Control for Submarine Chambers	M. Klein	1,131,712	Mar. 16, 1915
Means for Conducting Submarine Warfare	C. M. Wheaton, dec'd	1,131,761	Mar. 16, 1915
Submarine Torpedo Boat	J. Barraja-Frauenfelder	1,134,940	Apr. 6, 1915
Submarine Boat	S. Lake	1,135,537	Apr. 13, 1915
Safety Device for Submarine or Submergible Boats	E. L. Peacock	1,143,131	June 15, 1915
Breakwater Shield for Submarine Boats	R. H. M. Robinson	1,146,958	July 20, 1915
Submarine Boat	S. Lake	1,149,373	Aug. 10, 1915
Automatic Air Intake for Submarines, etc.	A. Hoar	1,151,540	Aug. 24, 1915
Submarine or Submersible Boat	E. L. Peacock	1,152,754	Sept. 7, 1915
Submarine Boat	L. Y. Spear	1,153,267	Sept. 14, 1915
Submarine Vessel	J. T. Parker	1,154,126	Sept. 21, 1915
Hull Construction of Submarine Boats	L. Y. Spear	1,154,215	Sept. 21, 1915
Submarine Boat	J. Barraja-Frauenfelder	1,158,883	Nov. 2, 1915

Submarine Vessel	C. Laurenti	1,161,484	Nov. 23, 1915
Submarine Boat	C. Regenbogen and H. Vogel	1,164,394	Dec. 14, 1915
Submarine Vessel	N. P. Nelson	1,165,535	Dec. 28, 1915
Device for Controlling the Water Ballast in Submergible Vessels	C. Sacerdoti	1,169,514	Jan. 25, 1916
Torpedo Launching Apparatus for Submarine Vessels	E. Schneider	1,166,940	Jan. 4, 1916
Hull for Submarines	M. F. Hay	1,169,640	Jan. 25, 1916
Submarine Boat	S. Lake	1,169,970	Feb. 1, 1916
Submarine Boat	H. E. Grieshaber	1,170,529	Feb. 8, 1916
Boat	H. W. Jacobs	1,173,431	Feb. 29, 1916
Submarine Torpedo Discharging Device	H. W. Hille	1,174,723	Mar. 7, 1916
Submarine Boat	J. Barraja-Frauenfelder	1,175,219	Mar. 14, 1916
Submarine Boat	S. Lake	1,180,263	Apr. 18, 1916
Submarine Boat	W. R. Mackland	1,180,861	Apr. 25, 1916
Submarine Boat	H. Technel	1,183,695	May 16, 1916
Submarine Boat	K. Voller	1,187,206	June 13, 1916
Submarine Boat	H. E. Grieshaber	1,187,522	June 20, 1916
Pressure Controlled Governing Apparatus	R. Janney	1,187,738	June 20, 1916
Submarine Buoyant Conveyance	A. Schrumpf	1,188,842	June 27, 1916
Submarine Boat	J. Barraja-Frauenfelder	1,190,210	July 4, 1916
Submarine Boat	W. F. Doherty, Jr.	1,192,172	July 25, 1916
Submarine Vessel	C. Laurenti	1,192,537	July 25, 1916
Submarine Vessel	C. A. Nallough	1,202,351	Oct. 24, 1916
Means for Destroying Vessels	F. V. Hagan	1,201,176	Oct. 10, 1916
Torpedo Tube for Submarines or the like	M. F. Hay and G. Ekama	1,204,353	Nov. 7, 1916
Water Tube Boiler for Submarines	G. C. Davison	1,209,678	Dec. 26, 1916
Submarine Steering Apparatus	A. Citroen	1,213,153	Jan. 23, 1917
Torpedo Launching Apparatus for Submarine Boats	S. Lake	1,215,387	Feb. 13, 1917
Apparatus for Moving Diving Rudder of Submarines	M. F. Hay and C. Ekama	1,216,564	Feb. 20, 1917
Safety Device for Submarine Boats	J. P. Ryan	1,219,667	Mar. 20, 1917
Cleaning Periscope Glasses	J. A. Steinmetz	1,222,156	Apr. 10, 1917
Submarine Warfare	J. A. Steinmetz	1,222,498	Apr. 10, 1917
Submarine	F. W. Schoenen	1,224,027	Apr. 24, 1917
Water Tube Boiler for Submarines	G. C. Davison	1,224,105	Apr. 24, 1917

II.—LIFE AND VESSEL SAVING DEVICES.

Subject	Patentee	Patent No.	Date Issued
Keel for Submarine Boats	W. Hammond	303,843	Aug. 19, 1884
Vessel for Submarine or Surface Navigation	V. B. De Souza	453,560	June 2, 1891
Submarine Locomotive	S. Lake	557,835	Apr. 7, 1896
Submarine Wrecking Boat	W. R. Hinsdale	575,890	Jan. 26, 1897
Combined Surface and Submarine Vessel	S. Lake	650,758	May 29, 1900
Submarine Apparatus	A. Von Hoffmann	657,218	Sept. 4, 1900
Submarine Apparatus	A. Von Hoffmann	664,152	Dec. 18, 1900
Submarine Boat	H. H. Morrell	726,085	Apr. 21, 1903
Means of Escape From Sunken Submarine and Similar Boats	F. T. Cable and L. Y. Spear	799,714	Sept. 19, 1905
Life Saving Boat	J. A. Marley	805,975	Nov. 28, 1905
Salvage Means for Submarine Boats	E. P. Dougherty	807,732	Dec. 19, 1905
Submarine Boat	W. J. O'Haire	814,869	Mar. 13, 1906
Marine Vessel	J. F. Gray	841,961	Jan. 22, 1907
Submarine Boat	R. H. Goldsborough	842,146	Jan. 22, 1907
Means of Escape From Sunken Submarines	E. A. Edney	845,623	Feb. 26, 1907
Submarine Structure	M. E. Pester	845,813	Mar. 5, 1907
Submarine Torpedo Boat	A. Elgar	846,736	Mar. 12, 1907
Means for Effecting Escape of Occupants From Sunken Vessel	I. Fripp	848,615	Mar. 26, 1907
Life Saving Device for Boats	J. Husser	855,890	June 4, 1907
Means for Recovering Submarine Boats	E. Oswald	856,096	June 4, 1907
Life Saving Apparatus for Submarine Boats	O. A. C. Oehmler	857,193	June 18, 1907
Signal Buoy and Lifting Device for Sunken Submarine Boats	E. L. Benson	861,795	July 30, 1907
Submarine Boat	W. J. O'Haire	867,294	Oct. 1, 1907
Means for Effecting Escape of Occupants From Sunken Vessels	D. A. Dempster	876,777	Jan. 14, 1908
Submarine Boat	P. Mott	883,040	Mar. 24, 1908
Boat Salvage Device	G. Salles	922,519	May 25, 1909
Navigating Turret for Submergible Vessels	S. Lake	925,706	June 22, 1909
Life Boat for Submarine Vessels	H. J. Walles	932,921	Aug. 31, 1909
Safety Device for Submarine Boats	W. A. Stevenson	952,379	Mar. 15, 1910
Life Saving Device for Submarine Boats	E. Labowsky	953,473	Mar. 29, 1910
Crew Saving and Submarine Salvaging Device	A. W. Reed	953,642	Mar. 29, 1910
Device for Raising Submarine Vessels	M. A. Laubeuf	955,414	Apr. 19, 1910
Submarine Boat	L. E. Goetz	960,537	June 7, 1910
Life Saving Attachment for Submarines	S. S. Peterson	974,516	Nov. 1, 1910
Life Saving Device	A. L. Bideaux	986,431	Mar. 14, 1911
Means for Indicating and Raising Sunken Boats	J. F. Shea	993,205	May 23, 1911
Submarine Boat	G. B. Yerton	994,552	June 6, 1911
Life Saving Device for Submarine Vessels	D. and J. Chappell	1,002,033	Aug. 29, 1911
Escape Device for Submarine Boats	J. Schnitter	1,007,145	Oct. 31, 1911

Signaling Apparatus for Submarine Boats	J. F. Shea	1,009,800	Nov. 28, 1911
Buoy Attachment for Submarine Boats	G. F. Keating	1,014,398	Jan. 9, 1912
Submarine Life Saving and Observation Tender	S. Pontiere	1,019,356	Mar. 5, 1912
Submarine Rescuing Apparatus	S. Roth	1,026,458	May 14, 1912
Submarine Signaling and Safety Device	C. Brown	1,028,989	June 11, 1912
Device for Raising Sunken Submarine Vessels	H. Aronsen	1,030,701	June 25, 1912
Wreck Raising Appliance	V. Buffulini and R. Rottenbacher	1,030,869	July 2, 1912
Life Saving Apparatus for Submarines	M. Reyngoudt	1,036,150	Aug. 20, 1912
Life Saving Attachment for Submarines	H. C. E. Lindtke	1,064,351	June 10, 1913
Life Saving Device	E. Topper	1,070,639	Aug. 19, 1913
Buoy for Submarine Boats	D. L. Cayo	1,071,393	Aug. 26, 1913
Signaling and Passenger Carrying Device for Vessels	A. Grossich	1,076,496	Oct. 21, 1913
Life Boat for Submarine Vessels	G. Hilgers	1,081,740	Dec. 16, 1913
Submarine Craft	K. Whiting	1,097,700	May 26, 1914
Submarine Boat	A. J. Griffin	1,103,958	July 21, 1914
Signal for Submarine Vessels	F. A. Lovegrove	1,113,799	Oct. 13, 1914
Apparatus for Raising Submarines and Submersibles	L. A. J. LeDuc	1,114,155	Oct. 20, 1914
Attachment for Submarine Boats	S. Danenhower	1,117,535	Nov. 17, 1914
Submarine Life Saving Device	C. W. Flint	1,117,826	Nov. 17, 1914
Safety Device for Submarine Boats	J. E. Conway	1,127,965	Feb. 9, 1915
Submarine Life Saving Appliance	C. G. Ullin	1,128,553	Feb. 16, 1915
Life Buoy for Submarine of Submergible Boats	H. Grieshaber	1,137,131	Apr. 27, 1915
Safety Appliance for Submarine Vessels	J. S. Fraser	1,139,409	May 11, 1915
Life Saving Apparatus	L. H. De Wyk	1,152,305	Aug. 31, 1915
Means of Escape Prom Submarines	W. J. Kenely	1,152,420	Sept. 7, 1915
Signal Device for Submarine Vessel	E. B. Walsh	1,153,277	Sept. 14, 1915
Means for Locating Submarine Boats	H. Lotz	1,154,522	Sept. 21, 1915
Safety Apparatus for Submarines	M. J. L. P. Bonard	1,156,970	Oct. 19, 1915
Anchor Cable Cutting Apparatus for Submarine Boats	J. Barraja-Frauenfelder	1,158,160	Oct. 26, 1915
Safety Float for Submarines	H. E. Keller and H. M. Siedschlag	1,159,683	Nov. 9, 1915
Submarine with Detachable Conning Tower for Life Saving Purposes	V. Cavallini	1,159,745	Nov. 9, 1915
Buoy for Submarines	W. H. Amberger	1,160,122	Nov. 16, 1915
Submarine	C. E. and H. E. Beck	1,166,651	Jan. 4, 1916
Submarine	R. Brunet	1,168,707	Jan. 18, 1916
Appliance for Vessels	E. F. Crane	1,169,128	Jan. 25, 1916
Escape for Submarines	E. L. Peacock	1,173,379	Feb. 29, 1916
Escape Apparatus for Submarine Boats	D. C. Marshall	1,173,726	Feb. 29, 1916
Submarine Vessel	C. Von Culin	1,175,848	Mar. 14, 1916
Apparatus for Locating and Raising Sunken Vessels	G. McKenzie	1,176,229	Mar. 21, 1916
Submarine	W. L. Walker	1,177,157	Mar. 28, 1916
Life Saving Device for Use in Submarines	H. F. Wilson	1,177,372	Mar. 28, 1916
Apparatus for Locating Sunken Vessels	J. F. and D. Neal	1,178,094	Apr. 4, 1916
Submarine Signaling and Salvage Apparatus	R. Okusa	1,178,407	Apr. 4, 1916
Signal Device for Submarine Boats	H. E. Nord	1,181,791	May 2, 1916

Subject	Patentee	Patent No.	Date Issued
Submarine Boat Attachment	W. H. Ramm	1,182,208	May 9, 1916
Marine Life Saving Apparatus	J. Konrad	1,183,078	May 16, 1916
Escape Apparatus for Submarines	R. Brunet	1,183,163	May 16, 1916
Wreck Indicating Device	G. W. Heath	1,184,180	May 23, 1916
Detachable Life Boat for Submarine Vessels	A. J. Morrison, Jr	1,184,981	May 30, 1916
Detachable Conning Tower and Life Buoy	C. Kaempfer	1,188,179	June 20, 1916
Device for Locating Sunken Vessels	C. P. Nyquist	1,188,308	June 20, 1916
Submarine Salvage Apparatus	D. W. Shea	1,188,333	June 20, 1916
Submarine Boat	G. A. Fredericks	1,188,436	June 27, 1916
Means to Raise Sunken Vessels	C. T. Bowers	1,188,888	June 27, 1916
Device for Locating and Raising Sunken Vessels	J. C. Paulson	1,189,864	July 4, 1916
Apparatus for Raising Sunken Vessels	R. W. Bryant and A. E. Moore	1,191,558	July 18, 1916
Automatic Signaling Device for Marine Vessels	P. Nelson	1,194,770	Aug. 15, 1916
Ship Locating Device	C. May	1,196,131	Aug. 29, 1916
Signaling or Indicating Buoy	C. M. Rollo	1,196,588	Aug. 29, 1916
Means for Floating or Raising Fluid Supported Objects	A. Musorofiti	1,197,650	Sept. 12, 1916
Life Saving Apparatus for Submarines	R. A. Willison	1,207,115	Dec. 5, 1916
Submarine Vessel	J. C. Buerke	1,207,851	Dec. 12, 1916
Vessel Locating and Salvaging Apparatus	J. M. Orr	1,208,362	Dec. 12, 1916
Submarine Life Boat	D. V. Reeves	1,210,840	Jan. 2, 1917
Submarine Life Saving and Observation Device	T. E. Burke and J. R. A. Lindholm	1,215,905	Feb. 13, 1917
Marker Buoy	L. A. Carter and L. J. Rector	1,215,908	Feb. 13, 1917
Life Boat for Submarines	W. Brown	1,221,282	Apr. 3, 1917
Vessel Indicating and Raising Apparatus	B. F. Nixon	1,223,606	Apr. 24, 1917

III.—TORPEDOES.

Subject	Patentee	Patent No.	Date Issued
Apparatus for Discharging Torpedoes Under Water	P. Plant	36,965	Nov. 18, 1862
Marine Torpedo	H. M. Nagel	39,162	July 7, 1863
Towing	J. D. Willoughby	48,124	June 6, 1865
Submarine Torpedo	W. R. King	102,556	May 3, 1870
Propulsion of Marine Torpedoes	J. A. Howell	121,052	Nov. 21, 1971
Submarine Torpedoes and Boats	H. J. Smith	134,493	Dec. 31, 1872
Torpedo Boats	H. P. Knapp	158,501	Jan. 5, 1875
Torpedo Guard	J. H. Fisher	197,262	Nov. 20, 1877
Electro-Magnetic Steering Apparatus for Submarine Boats	J. L. Lay	198,127	Dec. 11, 1877
Torpedo Boat	J. L. Lay	211,301	Jan. 14, 1879
Torpedo Boat	J. L. Lay	211,302	Jan. 14, 1879
Torpedo Boat	J. L. Lay	211,303	Jan. 14, 1879
Valves for Torpedo Boats	J. L. Lay	217,120	July 1, 1879
Marine Torpedo	J. H. McLean	222,718	Dec. 16, 1879
Torpedo Boat	W. H. Mallory	223,855	Jan. 27, 1880
Marine Torpedo	M. Coloney	225,465	Mar. 16, 1880
Torpedo Boat	G. W. McMullen	227,637	May 18, 1880
Submarine Torpedo Boat	M. A. Hardy	243,888	July 5, 1881
Submarine Torpedo Boat	M. Hubbe	243,907	July 5, 1881
Means for Adapting Submarine Guns to Ships	J. Ericsson	245,364	Aug. 9, 1881
Submarine Torpedo Boat	G. H. Reynolds	245,864	Aug. 16, 1881
Torpedo Boat	G. H. Reynolds	246,415	Aug. 30, 1881
Torpedo Guard for Vessels	W. H. Walker	247,717	Sept. 27, 1881
Gas Expansion Chamber for Torpedo Boats	G. E. Haight	250,144	Nov. 29, 1881
Torpedo Boat	G. H. Reynolds	251,288	Dec. 20, 1881
Submarine Torpedo	M. Hubbe and M. A. Hardy	255,386	Mar. 21, 1882
Submarine Torpedo Gun	P. Brotherhood	261,662	July 25, 1882
Submarine Torpedo	H. F. Hicks	263,407	Aug. 29, 1882
Apparatus for Launching Torpedoes	P. Brotherhood	261,805	July 25, 1882
Marine Torpedo	J. A. Howell	311,325	Jan. 27, 1885
Apparatus for Launching Torpedoes	J. A. Howell	311,326	Jan. 27, 1885
Torpedo	T. Nordenfelt	312,579	Feb. 17, 1885
Torpedo Boat Propelled and Steered by Electricity	W. S. Sims	319,633	June 9, 1885
Apparatus for Protecting Vessels Against Torpedo Attacks	A. C. Koerner	338,241	Mar. 16, 1886
Submarine Torpedo Boat	G. E. Haight and W. H. Wood	339,096	Mar. 30, 1886
Marine Torpedo	S. H. Nealy	356,471	Mar. 1, 1887
Electric Torpedo Boat	H. M. Bennett	359,313	Mar. 15, 1887
Safety Mechanism for Torpedo Tubes	E. Kaselowsky	361,066	Apr. 12, 1887

Automatic Brake Apparatus for Torpedoes	E. Kaselowsky	361,525	Apr. 19, 1887
Apparatus for Launching Torpedoes	E. Kaselowsky	361,526	Apr. 19, 1887
Torpedo Boat	A. P. S. Miller	364,364	June 7, 1887
Manufacture of Torpedo Nets and Means for Attaching and Working Them	W. M. Bullivant	379,854	Mar. 20, 1888
Auto-Mobile Torpedo	M. E. Hall	387,353	Aug. 7, 1888
Means for Operating Marine Torpedoes	J. O'Kelly and B. A. Collins	388,299	Aug. 21, 1888
Submarine Torpedo Boat	H. P. Griswold	388,862	Sept. 4, 1888
Net Cutting Attachment for Torpedoes	S. A. Brown	401,773	Apr. 23, 1889
Auto-Mobile Torpedo	A. G. Von Buonaccorsi Di Pistoja	413,113	Oct. 15, 1889
Self-Propelling Torpedo	A. G. Von Buonaccorsi Di Pistoja	413,114	Oct. 15, 1889
Automatic Sinking Valve for Submarine Torpedoes	A. G. Von Buonaccorsi Di Pistoja	413,115	Oct. 15, 1889
Auto-Mobile Torpedo	A. G. Von Buonaccorsi Di Pistoja	413,585	Oct. 22, 1889
Torpedo Launching Gear	H. P. Elwell	414,386	Nov. 5, 1889
Firing Mechanism for Torpedoes	A. G. Von Buonaccorsi Di Pistoja	417,206	Dec. 10, 1889
Torpedo and Gun Boat	R. J. Gatling	424,288	Mar. 25, 1890
Mechanism for Controlling Torpedoes	G. R. Murphy	442,327	Dec. 9, 1890
Marine Torpedo	N. J. Halpine	453,861	June 9, 1891
Universal Torpedo Dropping Gear	C. Dann	455,742	July 14, 1891
Marine Torpedo	A. W. Savage	456,278	July 21, 1891
Marine Torpedo	A. W. Savage	456,524	July 21, 1891
Marine Torpedo	J. A. Howell	458,677	Sept. 1, 1891
Torpedo Boat	J. Ambrose	464,909	Dec. 8, 1891
Torpedo Defense	W. P. Bullivant	466,976	Jan. 12, 1892
Apparatus for Launching Fish Torpedoes	J. B. G. A. Canet	470,286	Mar. 8, 1892
Torpedo Tube Mounting	J. B. G. A. Canet	470,288	Mar. 8, 1892
Torpedo Launching Apparatus	A. Scheibel	472,553	Apr. 12, 1892
Depth Regulating Mechanism for Marine Torpedoes and Other Vessels	F. H. Paine	478,813	July 12, 1892
Torpedo Launching Apparatus	J. A. Howell	484,658	Oct. 18, 1892
Torpedo	S. B. Mower and G. E. Haight	498,183	May 23, 1893
Torpedo Launching Tube	J. B. G. A. Canet	514,810	Feb. 13, 1894
Launching Tube for Torpedoes	E. W. Lloyd and C. W. Hutchinson	517,390	Mar. 27, 1894
Apparatus for Ejecting or Launching Torpedoes	S. Drzewiecki	542,876	July 16, 1895
Apparatus for Launching Torpedoes	E. W. Lloyd and C. W. Hutchinson	548,374	Oct. 22, 1895
Marine Torpedo	L. F. Johnson, W. J. Slacke and H. Lacy	559,711	May 5, 1896
Steering Mechanism for Torpedoes	L. Obry	562,235	June 16, 1896
Steering Apparatus for Torpedoes	E. Kaselowsky	591,768	Oct. 12, 1897
Automatic Steering Device for Torpedoes	E. Kaselowsky	607,440	July 19, 1898

Subject	Patentee	Patent No.	Date Issued
Torpedo Launching Apparatus	J. Whitehead	608,814	Aug. 9, 1898
Device for Starting Torpedoes	L. Obry	621,364	Mar. 21, 1899
Torpedo Launching Apparatus	S. Drzewiecki	621,640	Mar. 21, 1899
Apparatus for Launching or Discharging Torpedoes	H. L. J. C. Turck	626,945	June 13, 1899
Torpedo Launching Apparatus	E. Kasolowsky	631,308	Aug. 22, 1899
Automobile Torpedo	T. E. Barrow	632,089	Aug. 29, 1899
Torpedo	T. W. Just	638,463	Dec. 5, 1899
Automobile Torpedo	H. Maxim	641,787	Jan. 23, 1900
Automatic Steering Device for Torpedoes	C. D. Haskins	661,520	Nov. 13, 1900
Torpedo	E. Kaselowsky	661,535	Nov. 13, 1900
Submersion Regulating Gear for Torpedoes	A. J. Van Stockum	670,041	Mar. 19, 1901
Autotorpedo	H. Shoemaker	680,505	Aug. 13, 1901
Apparatus for Putting Torpedoes on Vessels	C. A. Morris	685,165	Oct. 22, 1901
Propulsion of Torpedoes, etc., by Compressed Air	F. M. Leavitt	693,872	Feb. 25, 1902
Firing Valve for Subsurface Expulsion Tubes	J. P. Holland	696,971	Apr. 8, 1902
Torpedo Launching Apparatus	J. Whitehead	697,906	Apr. 15, 1902
Torpedo Launching Apparatus	S. J. J. Drzewiecki	708,031	Sept. 2, 1902
Torpedo Boat	P. W. Brady	713,198	Nov. 11, 1902
Submerged Broadside Apparatus for Discharging Torpedoes	G. Hovos	713,985	Nov. 18, 1902
Steering Apparatus for Torpedoes	J. Borresen	716,517	Dec. 23, 1902
Apparatus for Handling Automobile Torpedoes	B. A. Fiske	717,563	Jan. 6, 1903
Torpedo	M. Fischhaber	726,796	Apr. 28, 1903
Compensating Device for Submarine or Submergible Boats	L. Y. Spear	739,735	Sept. 22, 1903
Gyroscopic Apparatus for Steering Torpedoes	F. M. Leavitt	741,683	Oct. 20, 1903
Means for Expelling Torpedoes	P. W. Brady	765,305	July 19, 1904
Safety Device for Automobile Torpedoes	G. E. Edgar	765,769	July 26, 1904
Steering Mechanism for Torpedoes	F. M. Leavitt	785,425	Mar. 21, 1905
Gyroscopic Control Apparatus	F. M. Leavitt	795,045	July 18, 1905
War Ship	J. Slonka	797,235	Aug. 15, 1905
Torpedo Launching Apparatus	A. E. Jones	801,719	Oct. 10, 1905
Boardside Under Water Torpedo Launching Apparatus	A. E. Jones	801,792	Oct. 10, 1905
Torpedo Handling Device for Vessels	G. M. Evans	803,218	Oct. 31, 1905
Automotor Torpedo	H. A. Noalhat and G. Fournier	807,195	Dec. 12, 1905
Torpedo Boat	T. J. Moriarty	807,482	Dec. 19, 1905
Marine Torpedo Guard	J. Neumaier	808,942	Jan. 2, 1906
Starting Valve for Automobile Torpedoes	F. M. Leavitt	814,055	Mar. 6, 1906
Gyroscope Spinning Device	F. M. Leavitt	814,969	Mar. 13, 1906
Torpedo Conveying and Launching Apparatus	T. H. Wheless	815,393	Mar. 20, 1906
Retarding Device for Automobile Torpedoes	F. M. Leavitt	816,019	Mar. 27, 1906
Means for Carrying and Handling Torpedoes on Vessels	T. J. Moriarty	818,390	Apr. 17, 1906
Automobile Torpedo	T. H. Wheless	818,987	Apr. 24, 1906

Valve Attachment for Torpedoes	E. Nichols	820,888	May 15, 1906
Revolving Torpedo Tube Cap	H. E. Grieshaber	820,925	May 15, 1906
Torpedo Expulsion Valve	J. Barraja-Frauenfelder	822,500	June 5, 1906
Automatic System for Balancing and Controlling Torpedoes	T. Gihon	825,881	July 10, 1906
Steering Apparatus for Automobile Torpodoes	F. M. Leavitt	839,161	Dec. 25, 1906
Automobile Torpedo	G. C. Davison	858,266	June 25, 1907
Means for Carrying and Handling Torpedoes on Vessels	T. J. Moriarty	868,613	Oct. 15, 1907
Torpedo Projecting Apparatus for Submarime and Submersible Vessels	H. Smulders	868,946	Oct. 22, 1907
Tandem Torpedo Tube	L. Y. Spear	871,453	Nov. 19, 1907
Starting Valve for Automobile Torpedoes	P. M. Leavitt	880,030	Feb. 25, 1908
Immersion Regulator	A. E. Jones	881,930	Mar. 17, 1908
Steering Apparatus for Torpedoes	A. J. VanStockum	882,982	Mar. 24, 1908
Means for Actuating Submersion Rudders for Torpedoes	A. E. Jones	883,028	Mar. 24, 1908
Torpedo Boat	E. A. Nilsen	883,664	Mar. 31, 1908
Torpedo Expulsion Tube	L. Y. Spear	888,544	May 26, 1908
Gyroscopic Steering Gear for Torpedoes	F. M. Leavitt	894,838	Aug. 4, 1908
Propelled Torpedo	H. Lacy	895,870	Aug. 11, 1908
Torpedo Boat	E. J. Kelley	896,921	Aug. 25, 1908
Immersion Regulator, Particularly Adapted for Torpedoes	A. E. Jones	899,304	Sept. 22, 1908
Shell or Casing for Standard Torpedoes	C. Davis	901,157	Oct. 13, 1908
Control Mechanism for Steering Apparatus	F. M. Leavitt	901,355	Oct. 20, 1908
Automatic Immersion Regulator for Submarines and Torpedoes	E. Schneider	904,093	Nov. 17, 1908
Torpedo	P. J. Hedlund	906,133	Dec. 8, 1908
Torpedo Launching Tube	A. E. Jones	908,270	Dec. 29, 1908
Torpedo	E. O'Toole	909,321	Jan. 12, 1909
Self-Propelled Torpedo	A. E. Jones	910,823	Jan. 26, 1909
Firing Means for Torpedoes	C. Davis	914,371	Mar. 2, 1909
Apparatus for Launching Torpedoes Under Water	A. E. Jones	916,164	Mar. 23, 1909
Self-Propelled Torpedo	A. E. Jones	917,449	Apr. 6, 1909
Torpedo Launching Device	S. Lake	925,707	June 22, 1909
Steering Mechanism for Torpedoes	P. M. Leavitt	925,710	June 22, 1909
Turbine Installation for Torpedoes	S. Z. De Ferranti	925,889	June 22, 1909
Air Heater for Compressed Air Engines	H. W. Shonnard and W. Dieter	927,434	July 6, 1909
Starting Device for the Compressed Air Motors of Self-Propelled Torpedoes	A. E. Jones	929,954	Aug. 3, 1909
Diving Gear for Submarine Boats	P. M. Leavitt	933,083	Sept. 7, 1909
Gas-Propelled Torpedo	I. N. Lewis	933,086	Sept. 7, 1909
Torpedo Tube	C. Laurenti	934,192	Sept. 14, 1909
Automobile Torpedo	H. Maxim	937,217	Oct. 19, 1909
Torpedo	M. Larsen	940,033	Nov. 16, 1909
Torpedo	J. Tasto	941,111	Nov. 23, 1909
Depth Regulating Mechanism for Moving Vessels	G. C. Davison	946,659	Jan. 18, 1910
Torpedo Launching Apparatus	S. Lake	949,325	Feb. 15, 1910
Torpedo and the like	K. O. Leon	952,451	Mar. 22, 1910
Apparatus for Discharging Excess Exhaust Gases from Motors Working on a Closed Cycle	G. F. Jaubert	953,146	Mar. 29, 1910
Exploder for Automobile Torpedoes	P. M. Leavitt	953,848	Apr. 5, 1910

Subject	Patentee	Patent No.	Date Issued
Torpedo	M. Glass	957,670	May 10, 1910
Attachment for Torpedo Tubes	J. Kimbrell	958,650	May 17, 1910
Torpedo	C. Davis	964,147	July 12, 1910
Torpedo	C. Davis	964,148	July 12, 1910
Gyroscopically Controlled Torpedo Firing Apparatus	C. Davis	972,064	Oct. 4, 1910
Self-Propelling Torpedo	H. W. Shonnard	973,141	Oct. 18, 1910
Automobile Torpedo	G. C. Davison	977,438	Dec. 6, 1910
Automobile Torpedo	W. M. Douglas	978,862	Dec. 20, 1910
Automobile Torpedo	H. G. Gillmor	980,243	Jan. 3, 1911
Gyroscopic Steering Apparatus	J. C. Waldron	983,467	Feb. 7, 1911
Submarine Broadside Torpedo Launching Tube	C. C. A. Fallenius	990,429	Apr. 25, 1911
Water Jet Marker for Torpedoes	G. P. Helfrich	995,138	June 13, 1911
Automobile Torpedo	A. E. Jones	996,412	June 27, 1911
Automobile Torpedo	J. M. O'Kelly	998,383	July 18, 1911
Automobile Torpedo	W. M. Douglas	998,475	July 18, 1911
Percussion Mechanism of Automobile Torpedoes	A. E. Jones	1,005,042	Oct. 3, 1911
Means for the Propulsion of Automobile Torpedoes	A. E. Jones	1,005,647	Oct. 10, 1911
Automobile Torpedo and Method of and Apparatus for its Propulsion	H. Maxim	1,014,014	Jan. 9, 1912
Automatic Controlling Means for Submarine Vessels	H. W. Shonnard	1,022,708	Apr. 9, 1912
Mechanical Air Device for Expelling Water or other Liquids from the Heads of Torpedoes to Aid in the Recovery Thereof After Firing	K. Whiting and J. B. Howell	1,023,907	Apr. 23, 1912
Torpedo Launching Tube	E. Schneider	1,024,424	Apr. 23, 1912
Automatic Steering Device for Torpedoes	H. W. Shonnard	1,030,134	June 18, 1912
Device for Cutting or Destroying Torpedo Nets	W. Golsteyn	1,031,997	July 9, 1912
Diving Gear for Torpedoes	F. M. Leavitt	1,033,810	July 30, 1912
Valve Control Mechanism for Gyroscopic Steering Gear	G. C. Davison	1,033,994	July 30, 1912
Gyroscopic Steering Mechanism for Torpedoes	G. C. Davison	1,033,995	July 30, 1912
Device for Destroying Torpedo Nets	W. Golsteyn	1,034,999	Aug. 6, 1912
Method of Fire Control for Torpedoes	H. W. Shonnard	1,035,647	Aug. 13, 1912
Submarine or Submergible Boat	E. L. Peacock	1,037,519	Sept. 3, 1912
Stopping Device for Automobile Torpedoes	F. M. Leavitt	1,044,543	Nov. 19, 1912
Means for Launching Torpedoes from the Sides of Ships	A. E. Jones	1,046,192	Dec. 3, 1912
Starting and Stopping Mechanism for Automobile Torpedoes	F. M. Leavitt	1,064,349	June 10, 1913
Exploder	F. M. Leavitt	1,058,594	July 29, 1193
Sinking Device for Automobile Torpedoes	F. M. Leavitt	1,076,235	Oct. 21, 1913
Submarine Torpedo	H. W. Shonnard	1,077,311	Nov. 4, 1913
Gyroscope Steering Gear	P. Hennig	1,077,344	Nov. 4, 1913
Steering Mechanism for Automobile Torpedoes	F. M. Leavitt	1,080,116	Dec. 2, 1913
Mechanism for Firing Torpedoes	T. S. Bailey	1,094,963	Apr. 28, 1914
Torpedo Launching Apparatus	T. McC. Gunn	1,097,623	May 26, 1914
Torpedo Launching Apparatus	T. McC. Gunn	1,097,624	May 26, 1914
Torpedo Launching Apparatus	T. McC. Gunn	1,097,625	May 26, 1914

Gyroscopic Steering Mechanism for Torpedoes	F. Zottich	1,098,074	May 26, 1914
Torpedo Steering Mechanism	J. J. Dallier	1,098,230	May 26, 1914
Guide Block for Torpedoes	W. J. Doolan	1,099,971	June 16, 1914
Broadside Firing Deck Torpedo Tube	M. F. Hay and F. Guhrauer	1,100,481	June 16, 1914
Broadside Firing Torpedo Tube	M. F. Hay and F. Guhrauer	1,100,676	June 16, 1914
Torpedo	G. P. Helfrich	1,117,843	Nov. 17, 1914
Indicator for Submarine Vessels	G. P. Smith	1,120,417	Dec. 8, 1914
Means for Automatically Steering Torpedoes or the like	K. O. Leon	1,121,563	Dec. 15, 1914
Torpedo Launching Apparatus	G. C. Davison	1,122,699	Dec. 29, 1914
Apparatus for Launching Torpedoes	G. C. Davison	1,122,700	Dec. 29, 1914
Gyroscopically Controlled Torpedo	E. F. Chandler	1,127,403	Feb. 9, 1915
Device for Saving Torpedoes	E. Durr	1,130,585	Mar. 2, 1915
Torpedo	G. P. Helfrich	1,133,282	Mar. 30, 1915
Torpedo and Other Submarine Apparatus	K. O. Leon	1,137,222	Apr. 27, 1915
Mine Catcher	J. K. Kendrick and J. W. Halterman	1,138,225	May 4, 1915
Mine Catcher	P. Zakit	1,138,525	May 4, 1915
Torpedo Guard	E. R. Trammell	1,139,239	May 11, 1915
Gyroscopic Steering Device	F. M. Leavitt	1,145,025	July 6, 1915
Diving Gear for Torpedoes	W. Dieter	1,145,355	July 6, 1915
Mine Raising Apparatus	J. C. Lobato	1,148,103	July 27, 1915
Automobile Torpedo	W. Dieter	1,148,155	July 27, 1915
Torpedo Launching Tube	W. Dieter	1,149,505	Aug. 10, 1915
Gyroscopic Steering Mechanism	W. Dieter	1,153,678	Sept. 14, 1915
Device to Prevent the Admission of Water into the Interior of the Hollow Shaft of a Torpedo's Propeller	C. Radiguer	1,154,206	Sept. 21, 1915
Dirigible Torpedo	M. P. Otto	1,154,427	Sept. 21, 1915
Gyroscopic Steering Mechanism for Automobile Torpedoes	F. W. Dodd	1,156,350	Oct. 12, 1915
Safety Mechanism for Operating the Muzzle and Breech Doors of Torpedo Firing Tubes	J. Barraja-Frauenfelder	1,158,884	Nov. 2, 1915
Torpedo Expulsion Tube	J. Barraja-Frauenfelder	1,161,182	Nov. 23, 1915
Torpedo	G. A. Knox	1,163,158	Dec. 7, 1915
Torpedo Mechanism Starter	G. A. Knox	1,163,159	Dec. 7, 1915
Gyroscopically Controlled Torpedo	H. W. Shonnard	1,163,606	Dec. 7, 1915
Automobile Torpedo	H. W. Shonnard	1,163,607	Dec. 7, 1915
Mounting for Torpedo Tubes or the like	H. H. Parsons	1,164,957	Dec. 21, 1915
Mine Destoryer	I. M. Ritter	1,174,026	Feb. 29, 1916
Automobile Torpedo	B. H. Scott	1,177,280	Mar. 28, 1916
Steering Apparatus for Torpedoes	F. M. Leavitt	1,179,440	Apr. 18, 1916
Steering Mechanism for Torpedoes	F. M. Leavitt	1,179,439	Apr. 18, 1916
Torpedo Tube	C. Laurenti	1,179,848	Apr. 18, 1916
Mine and Submarine Destroyer	W. H. Norfolk	1,181,339	May 2, 1916

Subject	Patentee	Patent No.	Date Issued
Method of Gyroscopic Control	H. Anschuetz-Kaumpfe	1,180,365	Apr. 25, 1916
Buoyancy Varying Device	D. F. Asbury	1,180,366	Apr. 25, 1916
Torpedo Construction	H. W. Shonnard	1,181,548	May 2, 1916
Torpedo Launching Apparatus	D. R. Battles	1,184,409	May 23, 1916
Shield for Battleships	J. Taggart	1,185,145	May 30, 1916
Ship Protector	H. J. Charles	1,185,983	June 6, 1916
Mine Destroying Means	E. T. Robeson	1,187,179	June 13, 1916
Gyroscopic Steering Device	O. K. Cazin	1,189,239	July 4, 1916
Diving Gear for Torpedoes	W. Dieter	1,190,871	July 11, 1916
Means for Preventing Attacks of Torpedoes and the like	K. O. Leon	1,195,042	Aug. 15, 1916
Ship's Torpedo Tube	B. Rosenbaum	1,195,739	Aug. 22, 1916
Apparatus for Launching Automobile Torpedoes	G. C. Davison	1,196,745	Aug. 29, 1916
Gyroscopic Steering Mechanism	P. M. Leavitt	1,197,134	Sept. 5, 1916
Mine Guards for Ships	W. H. Baker	1,200,068	Oct. 3, 1916
Mine or Torpedo Guard for Vessels	W. H. Baker	1,200,069	Oct. 3, 1916
Torpedo Guard	T. M. Daniels	1,202,810	Oct. 31, 1916
Gyroscopic Steering Mechanism	W. Dieter	1,204,852	Nov. 14, 1916
Mine Sweeper	M. W. Green	1,206,432	Nov. 28, 1916
Torpedo Construction	H. W. Shonnard	1,207,333	Dec. 5, 1916
Torpedo Tube	C. Laurenti	1,209,729	Dec. 26, 1916
Torpedo	A. Extrand	1,212,468	Jan. 16, 1917
Torpedo Catcher and Mine Destroyer	H. O'Grady	1,217,812	Feb. 27, 1917
Torpedo Guard for Ships	A. Carlson and F. A. Swanson	1,218,371	Mar. 6, 1917
Torpedo and Submarine Guard	J. A. Steinmetz	1,219,879	Mar. 20, 1917
Mine Catcher and Ship Protector	M. Feller	1,219,929	Mar. 20, 1917
Automatically Controlled Floating Mine	L. A. Bockstahler	1,221,964	Apr. 10, 1917
Torpedo Firing Control and Signal Apparatus for Torpedo Boats	E. G. Gallagher	1,222,198	Apr. 10, 1917
Aeromarine Torpedo	J. A. Steinmetz	1,223,212	Apr. 17, 1917
Torpedo Shield for Vessels	J. O. Travelstead	1,223,536	Apr. 24, 1917
Ship Protecting Shield	W. A. Shephard	1,223,627	Apr. 24, 1917
Submarine Vessel	A. J. Van Stockum	1,223,747	Apr. 24, 1917

BOOK REVIEWS.

THE SOLDIERS' ENGLISH AND FRENCH CONVERSATION BOOK, by WALTER M. GALLICHAN. Size 5¼ x 3 inches, pages, 128. J. B. LIPPINCOTT COMPANY, Philadelphia. Price 30 cents. A very carefully compiled book containing a great number of words, phrases and sentences selected especially for the use of the soldier. Not only is the spelling of the French given, but the pronunciation is clearly and accurately shown. It is a convenient size for the pocket, very clearly printed, and can not fail to be of great help to any English-speaking soldier in France or Belgium.

ASSOCIATION NOTES.

On May 25, 1917, Lieutenant Commander J. O. Richardson, U. S. Navy, tendered his resignation as Secretary-Treasurer of the Society due to his detachment from duty at Washington, effective May 31, 1917. The Council accepted the resignation and elected Lieutenant Commander F. W. Sterling, U. S. Navy, Retired, Secretary-Treasurer for the remainder of the current year.

On June 14, 1917, the Council directed the purchase of U. S. Liberty Bonds to the value of $1,000.

The following members and associates have joined the Society since the publication of the last Journal.

MEMBERS.

Brown, H., Ensign, Naval Reserve.
Burbank, J. A., Ensign, Naval Reserve.
Glass, H. De H., Ensign, Naval Reserve.
Hart, H. P., Ensign, Naval Reserve.
Hussey, George F., Jr., Ensign, U. S. Navy.
Keller, Jay H., Ensign, Naval Reserve.
Saxe, A. C., Ensign, Naval Reserve.
Slaymaker, W. W., Ensign, Naval Reserve.
Sturtevant, E. R., Ensign, Naval Reserve.
Westbrook, C. R., Ensign, Naval Reserve.

ASSOCIATE.

Kingsbury, Albert, 945 Oliver Building, Pittsburgh, Pa.